



Proceedings

ASPE 2015 Annual Meeting

November 1-6, 2015

Hyatt Regency Austin
Austin, Texas, USA

American Society for

Precision Engineering

P.O. Box 10826, Raleigh, NC 27605-0826
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Proceedings of:

The Thirtieth Annual Meeting of

The American Society for

Precision Engineering

November 1 - November 6, 2015
Hyatt Regency Austin
Austin, Texas, USA

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology – and to represent all facets from research to application.

The Annual Meeting has evolved into an international forum for the exchange of ideas and presentation of research results relating to precision engineering, metrology, controls, and system integration. Precision engineers and scientists from private industry, government laboratories, and universities meet to learn about the latest developments and to exchange ideas about the future directions of these technologies.

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Preface

This book comprises the proceedings of the 30th ASPE Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

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ASPE gratefully acknowledges the time and the effort of the organizing and technical program committee to bring the precision engineering community this program.

Contents

Preface	ii
Organizing Committee	iii
Table of Contents	iv
ASPE Lifetime Achievement Award	v
ASPE Distinguished Service Award	vi
2015 Scholarship Recipients and Sponsors	vii
ASPE Student Scholarships	viii
Sustaining Corporate Sponsors	ix
Corporate Sponsors	x
Technical Paper Index	1 - 11
Technical Papers	15 - 603
Author Index	605 - 608

ASPE Lifetime Achievement Award



Thomas A. Dow
North Carolina State University

The ASPE Lifetime Achievement Award is presented to individuals who have made significant contributions to the field of Precision Engineering. Dr. Thomas A. Dow is regarded worldwide as a founding father, a leading spokesperson and a top expert in the field of precision engineering.

After receiving his PhD degree from Northwestern University in 1972, Dr. Dow joined the Tribology Section of Battelle Columbus Laboratories and worked there for ten years. His seminal publications and ASME-published text on the mechanics of lubrication and wear in sliding contacts lead to the ASME Lubrication Division Newkirk Award in 1976 for contributions to metrology. These activities were his entry points into what was then the brand new field of precision engineering.

Dr. Dow joined the faculty at North Carolina State University in 1982 and established the first national academic research center, Precision Engineering Center, dedicated to precision engineering research and education. Dr. Dow's most notable early work was the development and demonstration of the fast-tool servo for machining non-rotationally symmetric optical surfaces. He was one of the first to integrate this concept into a diamond turning machine and the technique has since become indispensable in many ultraprecision applications. His work on servomechanisms and force transduction in diamond machining laid the foundation for his subsequent analytical, numerical, and experimental research on mechanisms of material removal at nanometer length scales. He is well known for developing models and measurement methods that relate achievable surface finish to diamond tool sharpness and nanoscale cutting forces. He developed a theory for ductile response in nominally brittle materials that led to a counterintuitive but now widely accepted model for ductile regime grinding. His landmark paper on that topic has been cited more than 500 times by others in the field. More recently he also developed models and experimental methods to study other new technologies such as vibration assisted machining and nanocoining.

Professor Dow's sustained commitment to precision engineering education has had a remarkable impact on the field. His inspiring pedagogy and rigorous educational approach has helped to launch the careers of dozens of aspiring precision engineers. He has always engaged all of his students in "active learning," well before that practice was popularized in engineering curricula. His students take field trips to organizations to see first-hand the challenges and opportunities associated with careers in precision engineering. His advisees generally feel more like apprentices than students. He works side by side with them in the lab, sharing insight and guiding progress. His broad practical mechanical engineering experience serves both as a model and as a resource.

Dr. Dow was one of the founders of the American Society for Precision Engineering and has acted as the Executive Director for the past 30 years.

ASPE Distinguished Service Award



Dan E. Luttrell
Kriterion, LLC

The Distinguished Service Award is presented to individuals who have dedicated significant time and expertise to the Society. The recipient of the 2015 Distinguished Service Award is Dan E. Luttrell, Kriterion, LLC.

Mr. Luttrell has over 27 years in the Precision Engineering field working for small companies and large corporations, and in technical positions as well as functional management positions. His technology focus has been the design, manufacture, and application of ultra-precision machine tools and measurement systems. In particular, he has extensive experience in the development of diamond turning machines, ultra-precision grinding machines, very high-precision thermal control systems, and optical metrology systems.

Mr. Luttrell has also held senior management positions in large and small companies. He was part of the startup of Precitech, Inc. in Keene, NH as its first Technical Director in the early 1990s. From there he held positions at two large corporations; first as a Senior Technical Specialist, 3M Company St. Paul, Minnesota, and then as the Core Technology Manager for Machine Research & Process Technology, Science & Technology Division, Corning Inc. More recently he was Vice President of Engineering, Moore Tool Company in Bridgeport, Connecticut. He is currently involved with the startup of Kriterion, LLC, which is a new company dedicated to development of innovative ultra-precision products for the measurement of highly aspheric and freeform optical surfaces.

Mr. Luttrell is a charter member of the ASPE and has twice served on the Board of Directors. He was Conference Chairman of the ASPE annual meeting in 1998, chaired the Business Forum for several years, has taught the Thermal Effects tutorial for 9 years, and developed the Project Management for Precision Engineers tutorial which he has taught for 2 years. He has also been a member of ASME (American Society of Mechanical Engineers) and IDSA (Industrial Design Society of America). Dan holds a BS Mechanical Engineering from the University of Tennessee, and an MS Mechanical Engineering from North Carolina State University where he was a research assistant in the Precision Engineering Lab under the direction of Dr. Thomas A. Dow.

2015 Scholarship Recipients

3M Student Scholarship



Congratulations to **Mr. Deokkyun Yoon**, University of Michigan, (Chinedum E. Okwudire, Advisor) for winning the 3M Student Scholarship Award. The 3M Student Scholarship includes a waiver of the Annual Meeting registration fee and 4 tutorial fees. It also includes an honorarium to cover travel and lodging expenses to the 30th ASPE Annual Meeting in Austin.

Carl Zeiss Student Scholarship



The winner of the 2015 Carl Zeiss Student Scholarship is Award is **Steven R. Gillmer**, University of Rochester, (Jonathan D. Ellis, Advisor). This award includes the ASPE Annual Meeting registration fee, 4 tutorial fees, travel expenses and lodging to the Annual Meeting. In addition, the scholarship will provide for travel and lodging to Maple Grove, Minnesota to visit Carl Zeiss Industrial Metrology. Mr. Gillmer is the third recipient of the Carl Zeiss Student Scholarship awarded through ASPE.

Lawrence Livermore National Laboratory Student Scholarships



Congratulations to **Massimiliano Ferrucci**, KU Leuven, (Wim DeWulf, Advisor) for winning the Lawrence Livermore National Laboratory (LLNL) ASPE Conference Scholarship. The LLNL Scholarship includes a waiver of the Annual Meeting registration fee and 4 tutorial fees. It also includes an honorarium to cover travel and lodging expenses to the 30th ASPE Annual Meeting in Austin.

Congratulations also go to **Lucas Valdez**, UNC-Charlotte, (John C. Ziegert, Advisor) for winning the Lawrence Livermore National Laboratory (LLNL) ASPE Conference Scholarship Honorable Mention Award. This scholarship provides a small stipend to help cover the cost of attending the 30th ASPE Annual Meeting in Austin.

ASPE Student Scholarships

ASPE is proud to award 2 student scholarships for the 30th ASPE Annual Meeting in Austin, Texas. This year's scholarship recipients are **Mr. Christopher M. Capocasale**, Georgia Institute of Technology, (Craig Forest, Advisor) and **Mrs. Nasim Habibi**, University of North Carolina–Charlotte, (Faramarz Farahi, Advisor). The ASPE Student Scholarships include a waiver of the Annual Meeting registration fee and 4 tutorial fees. It also includes an honorarium to cover travel and lodging expenses to the 30th ASPE Annual Meeting in Austin.

ASPE is able to award student scholarships because of the generous contributions from organizations, as well as from many ASPE Members and ASPE supporters. ASPE believes the education and support of the next generation of precision engineers is crucial to the advancement of the field.

Special thanks to the following corporate sponsors for the 2015 ASPE Student Scholarships:

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Technical & Poster Sessions

The technical program of the 2015 ASPE Annual Meeting contains over 120 papers on precision engineering advances. Papers that lent themselves to significant verbal interaction, concise but important discoveries, or strongly visual or tactile subjects have been selected for the poster presentation. Authors will be in attendance to discuss their work during both poster sessions on Tuesday, November 3 from 3:30 pm -5:00pm and Wednesday, November 4 from 3:30 pm -5:00pm

Session I and Welcome Remarks

Precision Roll to Roll Machine Design and Controls

Tuesday, November 3, 2015, 8:30 AM - 10:00 AM

Session Chair: Anthony W. Gatica (M3 Design, Inc.) and
Stuart T. Smith (University of North Carolina –Charlotte)

- 1. Machine Concepts and Developments in Creating a Precision Reel-to-Reel Film Fabrication System**
Shore, P.; Comley, P. D.; Morantz, P.; Allsop, J. (Cranfield University);
Walker, N. (Ixscient Limited); Li, S. (Microsharp Innovations Limited) 15
- 2. Equipment for Atmospheric Atomic Layer Deposition in Roll-to-Roll Processes**
Knaapen, R. J. (VDL Enabling Technologies Group) 19
- 3. Roll-to-Roll Graphene Transfer: Machine Design and Potential Challenges**
Xin, H.; Li, W. (University of Texas at Austin) 23
- 4. Control Strategy for Flexure-Based Precision Roll-to-Roll Machine**
Wang, D.; Zhou, X.; Chen, S.-C. (The Chinese University of Hong Kong) 27

Session II

Precision Roll to Roll Manufacturing and Inspection

Tuesday, November 3, 2015, 10:30 AM - 12:00 PM

Session Chair: Paul Shore (Cranfield University) and James M. Nelson (3M Company)

- 1. Precision Engineering of Multi-Layer Optical Films**
Hebrink, T. J.; Kivel, E. J. (3M Company) 32
- 2. Fabrication of Fine Pitch Vee-Groove Patterns on Drum Masters for Roll to Roll BEF Fabrication**
Morgan, C. J. (Moore Nanotechnology Systems, LLC) 37
- 3. Large-Area, High-Performance Functional Metrology System for Nanophotonic Device Manufacturing**
Gawlik, B.; Sreenivasan, S. V. (The University of Texas at Austin);
LaBrake, D. (Canon Nanotechnologies, Inc.) 42

4. Flexible Glass Substrates for Roll to Roll Manufacturing Garner, S.; Lin, J. C.; Kuo, K. T.; Tseng, P. L.; Lin, S. M.; Huang, M. H.; Li, X.; Smith, R. (Corning Incorporated); Chou, T. S.; Hsieh, C. W.; Wang, C.-Y.; Yang, S.; Wang, K. G.; Lin, H.-Y. (Industrial Technology Research Institute, (ITRI)).	48
5. Material Dispensing Options for Roll-to-Roll UV Imprint Lithography Singhal, S.; Abed, O.; Grigas, M. A.; Sreenivasan, S. V. (The University of Texas at Austin).	52

Session III

Precision Manufacturing

Wednesday, November 4, 2015, 8:30 AM - 10:00 AM

Session Chair: Alex Sohn (Oculus) and
John C. Ziegert (University of North Carolina–Charlotte)

1. Ultraprecision High Speed Cutting Brinksmeier, E.; Preuss, W. R.; Riemer, O.; Rentsch, R. (University of Bremen).	58
2. Cutting Force Measurement During Diamond Flycutting of Chalcogenide Glass Troutman, J. R.; Owen, J. D.; Davies, M. A.; Schmitz, T. L. (University of North Carolina–Charlotte)	64
3. Diamond Tool Wear When Machining Ferrous Materials Dow, T. A.; Suit, B.; Hesler, G. S. (North Carolina State University)	68
4. Process Damping Identification for Multiple Degree of Freedom Turning Operations Tyler, C.; Schmitz, T. L. (University of North Carolina–Charlotte)	74
5. Time Domain Modeling of Compliant Workpiece Milling Rubeo, M. A.; Schmitz, T. L. (University of North Carolina–Charlotte)	80

Session IV

Precision Mechanical Design

Wednesday, November 4, 2015, 10:30 AM - 12:00 PM

Session Chair: Mark T. Kosmowski (Electro Scientific Industries, Inc.) and
Mark A. Stocker (Cranfield Precision, Division of Fives Landis Ltd.)

1. Parallel Laser Machining and Metal Sintering Based on Temporal Focusing Wang, D.; Zhang, D.; Chen, S.-C. (The Chinese University of Hong Kong)	86
2. Using Controlled Thermal Expansion for Tool Alignment in Ultra-Precision Milling Schönemann, L.; Riemer, O.; Brinksmeier, E. (University of Bremen).	91
3. System Behaviour of a Multiple Overconstrained Compliant Four-Bar Mechanism van de Sande, W. W. P. J.; Aarts, R. G. K. M.; Brouwer, D. M. (University of Twente).	95
4. Study of Soft Tissue Cutting Based on a Precision Vibrating Blade Microtome Wang, J.; Zhou, X.; Chen, S.-C. (The Chinese University of Hong Kong)	100
5. Design of Constraint Systems Based on the Rayleigh Quotient and Minimax Properties of Eigenvalues Nayfeh, S. S.; Verdirame, J. M. (Equilibra Design Group)	105

Session V

Precision Metrology

Wednesday, November 4, 2015, 1:30 PM - 3:00 PM

Session Chair: Vivek G. Badami (Zygo Corporation – AMETEK) and
Robert D. Grejda (Corning Tropol Corporation)

- 1. Is One Step Height Enough?**
Leach, R. K. (University of Nottingham) 110
- 2. Three Dimensional Digital Holographic Aperture Synthesis for Rapid and Highly Accurate Large Volume Metrology**
Crouch, S.; Kaylor, B.; Reibel, R. R. (Bridger Photonics, Inc.);
Barber, Z. (Montana State University). 114
- 3. Systems-Level Integration for Focused Beam Scatterometry**
Gillmer, S. R.; Head, S. T.; Theisen, M. J.; Brown, T. G.; Ellis, J. D. (University of Rochester) 120
- 4. Design and Calibration of an Artifact for Evaluating Laser Scanning Articulating Arm CMMs Used for Measuring Complex Non-Concurrent Surfaces**
Lee, V.; Phillips, S. D.; Shakarji, C. M. (National Institute of Standards & Technology);
Hosto, J. J.; Huber, J. M.; Gillich, B. J. (U. S. Army Aberdeen Test Center). 126
- 5. Design of a Nanometer-Accurate Form Measurement Machine for Segmented Large Telescope Mirrors**
Bos, A.; Rosielle, P. C. J. N.; Steinbuch (Eindhoven University of Technology);
Henselmans, R. (NTS-Group) 131

Session VI

Control of Precision Mechatronic Systems

Thursday, November 5, 2015, 8:30 AM - 10:00 AM

Session Chair: Stephen J. Ludwick (Aerotech, Inc.) and
Antonius T. Peijnenburg (VDL Enabling Technologies Group)

- 1. Topology Optimization for the Conceptual Design of Precision Mechatronic Systems**
van der Veen, G.; Langelaar, M.; van Keulen, F. (Delft University of Technology);
van der Meulen, S.; Aangenent, W. (ASML) Laro, D. A. H. (MI-Partners). 136
- 2. Polymer Damper Technology for Improved System Dynamics**
Wullms, P.; Ruijl, T. A. M. (MI-Partners) 142
- 3. A Novel Approach for Reducing the Settling Time of Roller Bearing Nanopositioning Stages using High Frequency Vibration**
Dong, X.; Zhang, X.; Okwudire, C. E. (University of Michigan) 148
- 4. Active Assist Device for Heat and Vibration Reduction in Precision Scanning Stages**
Yoon, D.; Okwudire, C. E. (University of Michigan) 154
- 5. An Overview of Advanced LIGO Combination of Passive and Active Vibration Isolation**
Matichard, F.; Mittleman, R. (Massachusetts Institute of Technology);
Shapiro, Brett N.; Lantz, B. T. (Stanford University); Robertson, N. A. (California
Institute of Technology); Radkins, H. (LIGO Hanford Observatory) 160

Session VII

Micro and Nano Technologies

Thursday, November 5, 2015, 10:30 AM - 12:00 PM

Session Chair: Craig R. Forest (Georgia Institute of Technology) and
Mark L. Schattenburg (Massachusetts Institute of Technology)

- 1. Flexure Design of Active Microarchitected Materials That Achieve Programmable Properties via Control**
Hopkins J. B. (University of California–Los Angeles) 166
- 2. Hybrid Additive and Microfabrication of an Advanced Micromirror Array**
Panas, R. M.; Jackson, J. A.; Uphaus, T. M.; Smith, W. L.; Harvey, C. (Lawrence Livermore National Laboratory); Hopkins J. B. (University of California–Los Angeles) 171
- 3. Sub-5nm Dual-Field Overlay Control in Jet and Flash Imprint Lithography using Topology Optimization**
Ajay, P.; Cherala, A.; Sreenivasan, S. V. (The University of Texas at Austin) 177
- 4. Carbon Nanotube Growth Force Detection on Multi-Axis MEMS Sensor with Integrated Microheater**
Ladner, I.; Duenner, A.; Cullinan, M. A. (The University of Texas at Austin) 183
- 5. Design of a MEMS-Based Tunable Graphene Resonator System with Precision Strain and Force Metrology**
Sun, G.; Ladner, I.; Cullinan, M. A. (The University of Texas at Austin) 188

Session VIII

Freeform Optics Design and Fabrication

Thursday, November 5, 2015, 1:30 PM - 3:00 PM

Session Chair: Eric S. Buice (Carl Zeiss Laser Optics GmbH) and
Christopher J. Evans (University of North Carolina–Charlotte)

- 1. Active Optics in the Manufacture of 8.4 m Mirrors for Telescopes**
Martin, H. M.; Cuerden, B. (University of Arizona) 194
- 2. Fabrication of an IR Telescope for a Balloon-Borne Space Mission**
Dow, T. A.; Furst, S. J.; Garrard, K. P. (North Carolina State University) 200
- 3. Productive Freeform Machining Using a Long Stroke Fast Tool System**
Niehaus, F.; Huttenhuis, S. (Schneider GmbH & Co. KG) 206

Educational Open Forum

Thursday, November 5, 2015, 12:00 PM - 1:30 PM

Session Chair: Byron R. Knapp (Professional Instruments Company)

- 1. The LEGO Watt Balance: A Simple Instrument to Demonstrate Precision Metrology**
Chao, L. S.; Schlamming, Pratt, J. R. (National Institute of Standards & Technology) 210

2. Microphone Squeal and Machining Chatter Schmitz, T. L. (University of North Carolina–Charlotte)	214
3. Carbon-graphite Gas Bearings for Turbomachinery Devitt, A. J. (New Way Air Bearings); Andres, L. S.; Jeung, S.-H. (Texas A&M University)	218

Poster Session

Tuesday, November 3, 2015, 3:30 PM - 5:00 PM

Wednesday, November 4, 2015, 3:30 PM - 5:00 PM

Advances in Precision Engineering and Nano Technologies

1. Compliant Pin Wafer Chucks for Yield Enhancement in Semiconductor Fabrication Ajay, P.; Westfahl, D.; Sreenivasan, S. V. (The University of Texas at Austin)	223
2. Study on the Influence Factors of Error Averaging in Hydrostatic Lead Screw Chen, Y.; Wang, T.; Zha, J.; Liu, L.; Jia, Q. (Xi'an Jiaotong University)	228
3. Applying Precision Engineering Principles to High Volume Product Development Gatica, A. W. (M3 Design, Inc.)	232
4. Development of Micro-Scale Bending Fatigue Test System for Real Time Observation and Accurate Load Measurement Hayakawa, N.; Tsuchiya, K. (The University of Tokyo); Fujisawa, R. (Mitsubishi Heavy Industries, Ltd.); Kakiuchi, T. (Gifu University)	237
5. The Motion of a 6-DOF Inchworm on Rough Surface Ishibashi, S.; Mitsuyoshi, Y.; Torii, A.; Doki, K.; Mototani, S. (Aichi Institute of Technology)	241
6. Tool Position Compensation for Machining a Micro-Prism Pattern on a Sinusoidal Surface by Shaping Je, T.-J.; Choi, H.-J.; Jeon, E.-C.; Jeong, J.-Y. (Korea Institute of Machinery and Materials (KIMM)); Park, E.-S. (University of Science & Technology)	245
7. Comparison of Various Trigger Touch Probes Maier, C.; Fritz, M.; Tobies, H. F. (KERN Microtechnik GmbH)	249
8. Micro-Laser Assisted Drilling of Single Crystal Silicon in Ductile Regime Mohammadi, H.; Patten, J. A. (Western Michigan University)	254
9. Study on Improving Machining Efficiency of Internal Magnetic Abrasive Finishing Process Muhamad, M. R.; Zou Y. (Utsunomiya University)	258
10. Evaluation of the Range Performance of Laser Scanners Using Non-planar Targets Rachakonda, P. K.; Muralikrishnan, B.; Shakarji, C. M.; Lee, V.; Sawyer, D. S. (National Institute of Standards & Technology)	262
11. Micro-Laser Assisted Machining: An Innovative Technology to Machine Optics Productively Ravindra, D. R.; Kode, S. K. (Micro-Laser Assisted Machining Technologies, LLC)	268

12. Balancing Concepts for High Speed Aerostatic Spindles Riemer, O.; Schönemann, L.; Doergeloh, T.; Beinhauer, A.; Foremny, E.; Brinksmeier, E.; Kuhfuß, B. (Universität Bremen)	273
13. Uncertainty of In-Process Roundness Measurement Using the Multi-Probe Error Separation Method Valdez, L.; Ziegert, J. C.; Schmidt, P. (University of North Carolina–Charlotte).	275

Control of Precision Mechatronics Systems

1. Force Estimation Method of Piezoelectric Actuator by Using Driving Current Furutani, K.; Inukai, R. (Toyota Technological Institute); Chee, S. K.; Yano, T. (MechanoTransformer); Higuchi, T. (University of Tokyo)	280
2. Straightness Control of a Long-Stroke, High-precision Scanner Stage Krishna, S. R.; Park, S.-H.; Jang, S.-D. (Samsung Electronics).	285
3. On-line Temperature Prediction of a Permanent Magnet Tubular Linear Motor for Control Parameter Tuning Li, F.; Ye, P.; Zhang, H. (Tsinghua University)	289
4. Dynamic Modeling of an H-Type Precision Gantry Including Base Motion Effects Ludwick, S. J. (Aerotech, Inc.)	293
5. Input Signal for Levitation Caused by Vertical Vibration of a Piezoelectric Actuator Torii, A.; Sone, S.; Doki, K.; Mototani, S. (Aichi Institute of Technology).	299
6. Interferometric Active Inertial Isolation for Extended Structures Watchi, J. J.; Collette, C. (Université Libre de Bruxelles); Artoos, Kurt (European Organization for Nuclear Research) Matichard, F. (Massachusetts Institute of Technology)	303
7. A Multi-Axis Motion Control Solution of Linear Ultrasonic Motors for Micro CNC Machine Tools Powered by Industrial Ethernet Technology (ETHERCAT) Xiao, W.; Huang, H.; Zhao, G. (Beihang University)	307
8. Active Assist Device for Heat and Vibration Reduction in Precision Scanning Stages Yoon, D.; Okwudire, C. E. (University of Michigan)	154

Control of Thermal Behavior of Precision Systems

1. Research on Improving the Thermal Stability of Hydrostatic Guideway Structure by Realizing the Homogenization of the Temperature Distribution Liu, X.; Chen, Y.; Jia, Q. (Xi'an Jiaotong University); Lan, J. (Shanghai Institute of Spaceflight Control Technology)	311
--	-----

Design of Precision Machines

1. Parallel Wires Driven Micro Manipulator Aoyama, H.; Senda, T. (University of Electro-Communications)	317
---	-----

2. Measurement of Sub-Micrometer Features in Borosilicate Glass Micropipettes Capocasale, C. M.; Stockslager, M. A.; Simon, M. D.; Li, Y.; McGruder, D. J.; Holst, G. L.; Forest, C. R. (Georgia Institute of Technology)	321
3. Design, Development and Testing of Precision Hydrostatic Rotary Table for Milling-Grinding Machine Chen, Y.; Zha, J.; Jia, Q.; Zhang, C.-X. (Xi'an Jiaotong University)	327
4. Modeling of Longitudinal-Bending Coupled Elliptical Vibration Cutting Tool Using Timoshenko Beam Theory Guo, P. (The Chinese University of Hong Kong)	331
5. Microscopic-Based Imaging for Precision Measurement of Micro Scale Flows Around Dynamically Oscillating Objects Kafashi, S.; Kelly, S.; Smith, S. T. (University of North Carolina – Charlotte); Eldredge, J. (University of California – Los Angeles)	336
6. Mobility Analysis of Flexure Systems Using Load Flow Visualization Krishnan, G.; Satheeshbabu, S. (University of Illinois at Urbana Champaign)	342
7. Design of Spindle Supported by Water Hydrostatic Bearings Nagasaka, K.; Hayashi, A.; Nakao, Y. (Kanagawa University)	348
8. Design of Compact Multi-DOF Ultrasonic Spherical Actuator Park, J.-H.; Jeong, J.-H.; Gweon, D.-G. (Korea Advanced Institute of Science and Technology (KAIST)); Kim, H.-Y. (Korea Institute of Industrial Technology)	353
9. New Identification Method for Dynamic Characteristics of Roller Guideway Based on Finite Element Model Sakai, Y.; Jiang, Z.; Yoshioka, H. (Tokyo Institute of Technology); Tanaka, T. (Nagoya University)	357
10. Design of a Suspended Matrix Sensor for Pressure Measurement Shi, H. (The Ohio State University)	363
11. Biosimulant Artifact with Embedded Calcium Alginate Bead Sensor for Robot Impact Safety Testing Shi, H.; Dagalakakis, N. G. (National Institute of Standards & Technology)	369

Education or Historical Perspectives on Precision Engineering

1. High Resolution Flexure-Based Tip-Tilt Table Knapp, B. R.; Arneson, D. A.; Oss, D. D. (Professional Instruments Company)	375
2. Testing the Earliest Sub-Microinch Spindles Liebers, M. J.; Arneson, D. A.; Oss, D. D. (Professional Instruments Company)	379

Interferometry and Optics

1. Oblique Incidence Stitching Interferometry for Manufacturing High Precision Flats Boland, R. J. (Zygo Ametek Ultra Precision Technologies)	383
---	-----

2. Computational Spectral Imaging in Chromatic Confocal Metrology	
Habibi, N.; Azari, M.; Farahi, F. (University of North Carolina – Charlotte); Abolbashari, M. (Optoniks, Inc.)	389
3. Surface Profile Measurement from an Oblique Direction Using a Change in Reflection Angle	
Kamiya, K.; Urata, Y.; Matsumoto, K.; Nomura, T. (Toyama Prefectural University); Tashiro, H. (Toyama University); Suzuki, S. (Nagano National College of Technology)	394
4. Novel Displacement Sensing Technique Utilizing Knife-Edge Diffraction	
Lee, C.-B. (Tennessee Technological University); Tarbutton, J. (University of South Carolina); Lee, S.-K. (Gwangju Institute of Science & Technology (GIST))	398
5. Improvement of a Periodic Error Compensation Algorithm Based on the Continuous Wavelet Transform	
Lu, C.; Tarbutton, J. (University of South Carolina); Lee, C.-B. (Tennessee Technological University); Ellis, J. D. (University of Rochester); Schmitz, T. L. (University of North Carolina – Charlotte).	403
6. Modular Versatile Imaging, Focusing & Illumination for Vision with High Stability and Accuracy	
Stevens, R.; Kusters, E.; Jacobs, R. (Settels Savenije van Amelsvoort)	407
7. Real-Time Periodic Error Correction for Heterodyne Interferometers Using Kalman Filters	
Wang, C.; Ellis, J. D. (University of Rochester).	409
8. Compact Six Degree-of-Freedom Interferometer for Precision Linear Stage Metrology	
Yu, X.; Gillmer, S. R.; Ellis, J. D. (University of Rochester); Howard, S. C.; Woody, S. (InsituTec, Inc.)	415

Manufacturing and Metrology of Freeform Surfaces

1. Surface Finish and 3D Optical Scanner Measurement Performance for Precision Engineering	
Dury, M. R.; Woodward, S. D.; Brown, S. B.; McCarthy, M. B. (National Physical Laboratory)	419
2. Influence of the Cutting Forces on the Structure of a Machine Tool CN-EMCO PC MILL 155-	
Merghache, S. M.; Ghernaout, A.; Cheikh, A. M. (Abou Bakr Belkaid University)	424
3. Performance Evaluation of a Laser Tracker Hand Held Touch Probe	
Muralikrishnan, B.; Blackburn, C.; Rachakonda, P. K.; Sawyer, D. S. (National Institute of Standards & Technology)	431
4. Suppression of Tool Wear in Ultra-Precision Diamond Turning of Steel by Nitriding	
Nagoshi, M.; Uda, Y.; Shimada, S. (Osaka Electro-Communication University); Honda, S. (Technology Research Institute of Osaka Prefecture)	435
5. Artifact Based Diamond Ball Mill Error Correction for Freeform Optics Manufacturing	
Owen, J. D.; Davies, M. A. (University of North Carolina – Charlotte)	440

6. 3 Axis Homodyne Displacement Measuring Interferometer Probe for Freeform Metrology	
Ricci, M. A.; Butler, S.; Wang, C.; Ellis, J. D. (University of Rochester)	445
7. Manufacturing Polymeric Micro Lenses and Lens Arrays by a Multiphase Printing Method	
Sun, R.; Li, L. (Washington State University)	451
8. Surface Shape Measuring Machine of Large Curved Surface Applying Improved Sequential Three-Point Method	
Tamagawa, T.; Uda, Y.; Shimada, S.; Imura, R. (Osaka Electro-Communication University).	457
9. Precise Non-Contact Center Thickness Metrology	
Thorpe, M. J.; Brasseur, J. K.; Roos, P. A. (Bridger Photonics, Inc.)	461
10. Understanding Articulating Arm Laser Line Scanners for Precision Engineering: Operator Usage Effects	
Woodward, S. D.; Dury, M. R.; Brown, S. B.; McCarthy, M. B. (National Physical Laboratory).	465

Micro and Nano Metrology and Measurement Uncertainty

1. Self-Calibration of the Non-Linear Length Deviation of a Linear Encoder by Using Two Reading Heads	
Bosmans, N.; Qian, J.; Reynaerts, D. (KU Leuven).	469
2. Experimanetal Verification of Alignment Related Errors in Position Metrology of Axes of Rotation	
Du, J. (Harbin Institute of Technology); Kang, W.; Bridges, P.; Miller, J. A. (University of North Carolina – Charlotte)	474
3. Passive Semiconductor Wafer Alignment Mechanism to Support In-line Atomic Force Microscope Metrology	
Duenner, A.; Cullinan, M. A. (The University of Texas at Austin)	480
4. Modelling the Effects of Detector Misalignments on the Measurement of a Cylindrical Array of Aluminum Spheres by X-ray Computed Tomography *	
Ferrucci, M.; McCarthy, M. B. (National Physical Laboratory); DeWulf, W. (KU Leuven)	
5. The Hybrid Measurement Technique for the PDGEs and PIGEs at the Identical Coordinate System	
Lee, J.-C.; Lee, H.-H.; Yang, S.-H. (Kyungpook National University).	485
6. Characterization of Directional Features for Precision Surfaces Using Curvelet	
Liu, M.; Cheung, C.-F.; Ren, M. (The Hong Kong Polytechnic University)	491
7. Errors in Surface Texture Measurements by Optical Method Caused by Non-Measured Points Presence	
Pawlus, P.; Reizer R. (University of Rzeszow).	497
8. Morphological Filtration of Stratified Textures	
Pawlus, P.; Reizer R. (University of Rzeszow); Łętocha, A. (The Institute of Advanced Manufacturing Technology).	503

****No Abstract Available***

- 9. Optimal Specimen Orientation in Cone-Beam X-Ray CT Systems (for Metrology)**
Villarraga-Gómez, H.; Smith, S. T. (University of North Carolina –Charlotte) 509
- 10. In-line, Wafer-Scale Inspection in Nano-Fabrication Systems**
Yao, T.-F.; Duennen, A.; Cullinan, M. A. (The University of Texas at Austin) 516

Nano-Positioning

- 1. A Novel Approach for Reducing the Settling Time of Roller Bearing Nanopositioning Stages Using High Frequency Vibration**
Dong, X.; Zhang, X.; Okwudire, C. E. (University of Michigan) 148
- 2. Proposal of a Novel Pressure Control Method for Air Cylinders Used in Hybrid Electric-Pneumatic Ultra-Precision Vertical Positioning Device**
Kato, T.; Hirakawa, T. (Fukuoka Institute of Technology) 522
- 3. Modeling and Optimization of MEMS Thermal Actuator**
Shi, H.; She, Y.; Su, H.-J. (The Ohio State University); Kim, Y.-S. (University of Maryland) 526
- 4. Design of a Parallel MEMS Positioning Stage with Decoupled Motion**
Shi, H.; She, Y.; (The Ohio State University); Kim, Y.-S. (University of Maryland) 532
- 5. Nanopositioning Stages Design for Four-Crystal Hard X-Ray Beam Split-and-Delay Line with Coherence Preservation**
Shu, D.; Shvyd'ko, Y.; Stoupin, S.; Anton, J.; Kearny, S.; Goetze, K. A. (Argonne National Laboratory) 534

Precision Manufacturing and Assembly Processes

- 1. Fast Calibration Tools and Strategy for Five-Axis Milling Machine**
Aguirre, G.; Pérez de Nanclares, A.; Urreta, H. (IK4-IDEKO) 538
- 2. Precision Manufacturing of Thermoplastic Elastomer Micro Rings**
Calaon, M.; Tosello, G.; Hansen, H. N. (Technical University of Denmark); Elsborg, R. (Ortofon A/S) 543
- 3. A Study of Factors Affecting Material Removal in Computer-controlled Bonnet Polishing**
Cheung, C.-F.; Cao, Z. C.; Ho, L.-T. (The Hong Kong Polytechnic University) 547
- 4. Milling Bifurcation at Extended Axial Depths of Cut**
Honeycutt, A.; Schmitz, T. L. (University of North Carolina –Charlotte) 551
- 5. Categorization and Review of Existing Micro-mirror Array Technologies**
Hopkins J. B.; Song, Y. (University of California –Los Angeles); Panas, R. M. (Lawrence Livermore National Laboratory) 555
- 6. Precision Machining of Yttria-Stabilized Tetragonal Zirconia Polycrystal by High-Speed Milling**
Ito, Y.; Kizaki, T.; Fujii, T.; Yuasa, Y.; Sugita, N.; Mitsuishi, M. (The University of Tokyo) 561
- 7. Quality Performance Test for Super Precision Spindle**
Kushnir, E.; Aguilar, A.; Clark, W. H. (Hardinge, Inc.) 567

8. Analytical Model for Thin Plate Dynamics	
Menezes, J.; Schmitz, T. L. (University of North Carolina–Charlotte); Kiran, K. (Suleyman Demirel University)	571
9. Multi-DOF Manipulator with Force Feedback for Target Assembly of HEDS Targets	
Panas, C.; Doane, D.; Felker, S. J.; Seugling, R. M.; Shaw, J. (Lawrence Livermore National Laboratory)	577
10. Tunable Dynamics Platform for Milling Experiments	
Ransom, T.; Honeycutt, A.; Schmitz, T. L. (University of North Carolina–Charlotte).	583
11. Internal Stresses in Plastic Optical Components Caused by Injection Molding and Injection Compression Molding	
Schmuetz, J.; Fadzil, L. (Beuth University of Applied Sciences)	587
12. Influence of Polymer Melt Properties in Precision Injection-Molding of Microfluidical and Optical Parts	
Schmuetz, J.; Heinrich, T. (Beuth University of Applied Sciences)	591
13. Fundamental Study on Metal Film Coating by Large-area EB Irradiation	
Takata, M.; Shinonaga, T.; Okada, A.; (Okayama University); Inoue, M.; Sano, S. (Sodick Co., Ltd.)	595

***Precision Supporting Semiconductor, MEMS and Photonics
Manufacturing and Packaging***

1. Design and Analysis of MEMS Cartwheel Hinge Joint	
Shi, H. (The Ohio State University); Kim, Y.-S. (University of Maryland)	599



Technical Papers

MACHINE CONCEPTS AND DEVELOPMENTS IN CREATING A PRECISION REEL TO REEL FILM FABRICATION SYSTEM

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INTRODUCTION

The size of features to be incorporated within film based products such as flexible displays are decreasing. In order to cost effectively produce film based electronic products it will be necessary to precisely handle and manipulate large scale (> 1 metre wide webs) films. Figure 1 indicates the present film production capability in regards throughput in metres² / second and the size of feature size in micrometres.

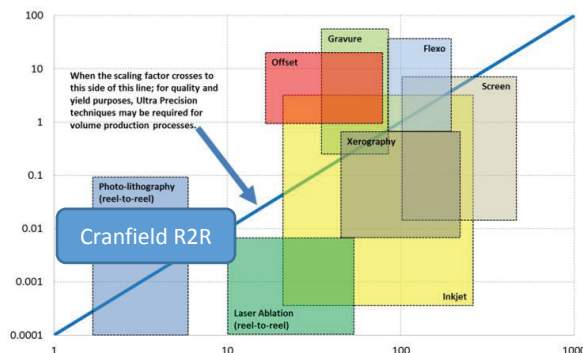


FIGURE 1. Film throughput (m²/second) versus feature size (micrometers) with indication of the Cranfield R2R target specification range

Furthermore the processing of the film substrates will include a number of processes of additive, subtractive and forming nature. These manufacturing processes will induce heat to the film, change its basic mechanical properties and consequently induce film substrate distortion. These “workpiece” issues are considered key to effective production of proposed and emerging products.

This paper introduces some of the machine concepts and developments of a large scale

R2R research platform under construction at Cranfield which is funded by UK's EPSRC Centre for Innovative Manufacturing in Ultra Precision. This R2R platform is in its construction and initial testing phases. An explanation of the key background, aims and objectives of this R2R system are provided along with an outline of the expected future research plan.

BACKGROUND

Since 1968 the Precision Engineering activity at Cranfield University has performed research, development and supply of ultra precision machines. Many machines that were created through this R&D were created to produce critical components of large science programmes. These machines have been widely reported in the literature [1,2]. Often the created machine tool technologies were subsequently incorporated into industrial production machines made at Cranfield. These production machines were key enabling machines of advanced production lines / manufacturing chains. The special purpose commercial machines were typically under tight industrial NDA's. Today the commercial products of Cranfield spin out companies' give evidence this ultra precision machine research contribution claim.

In 2007 a consortium led by Cranfield University established the EPSRC Integrated Knowledge Centre in Ultra Precision and Structured Surfaces. This EPSRC Centre, with support of Welsh Government created the UPS² laboratory. This ultra precision production facility is situated in North Wales at the OptIC Technium. It is operated through Cranfield and its spin out company Loxham Precision Limited. Over the last 5 years UPS² has been producing large

scale micro-structured drum moulds for production of high value passive optical films, fabric texturing systems and active electronic substrates. An example is shown in Figure 2. [3,4].



FIGURE 2. UPS² produced large scale micro-structured drum roll (1.4m long by 0.4m diameter)

In 2011, Cranfield University and the University of Cambridge, together with support of the National Physical Laboratory, secured funding for the EPSRC Centre for Innovative Manufacturing in Ultra Precision. A major task of this centre was identified as the realization of the R2R film processing research platform [5]. This research platform was targeted to provide three key demands:

1. To establish an effective R2R research platform to perform production research for a variety of future active film based products.
2. To provide a large scale 1.5 m wide film fabrication facility capable of producing high value passive films at high processing rates.
3. Establish an ultra-precision manufacturing supply chain that can better appreciate, and support, the needs of the emerging film based product researchers / developers. Many of which are based in the UK.

CRANFIELD R2R RESEARCH PLATFORM

The Cranfield R2R research platform incorporates a number of necessary processing technologies that lead to difficult, antagonistic, technologies for effective fine feature creation with high accuracy over large areas at high throughput. Rapid production of ultra precise fine featured films will require high temperature processes together with highly susceptible

substrates. Consequently, the traditional ultra precision approaches of isolating major heat generation systems are not entirely possible. The Cranfield R2R research platform is specified to eventually incorporate a number of processes these include:

Instrumented film control positioning
Roll embossing – programmable curing
Slot die deposition technology
Ink jet printing - laser processing
Multi film lamination

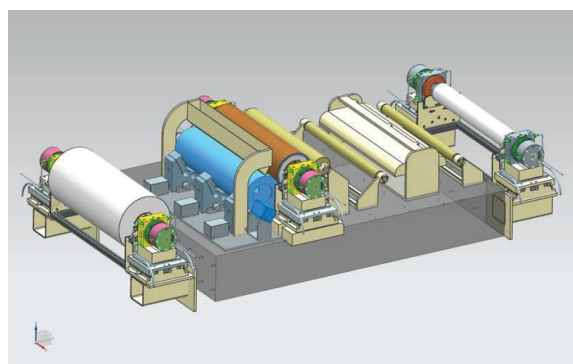


FIGURE 3. Cranfield R2R schematic outline

Development of the Cranfield R2R platform remains in progress [6]. Some of the key techniques and subsystem developments are summarised in this overview paper.

Instrumented film control positioning

In order to position and repeatedly reposition large scale films it requires an understanding of the dimensional state of the film. To accommodate these demands an initial instrumenting operation of films has been conceived.

As an initialising process substrates are modified by introducing a group of fine feature measurement scales into the film. These grating features enable the film position, reposition and dimensional change to be measured during R2R processing. An example “multiple grating” embossing roll mounted on specially designed ultra precision hydrostatic bearing units is shown in Figure 4.

This ability to induce fine gratings together with ultra precision positioning spindles offer sub-micrometre position measurement accuracy of films and a micrometer range positioning capability.

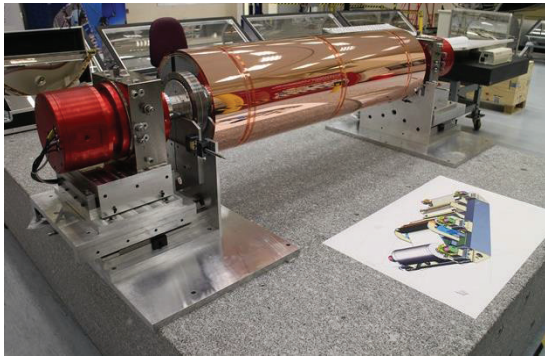


FIGURE 4. Grating inducing embossing roll mounted on high precision spindle units

Hydrostatic bearing positioning units

Fluid film bearing based positioning units are highly compact with low mass, low thermal capacity and with high diffusivity. They offer stiff sub – micrometre accuracy of axes of rotation which yields large drum roll run-out levels in the micrometre region.

The bearing units are constructed from aluminium and employ diamond turnable coatings having low friction and resistant wear characteristics. The coating technology, derived from the Formula1 Motorsport sector was developed by Poeton Coatings of Gloucestershire, England.

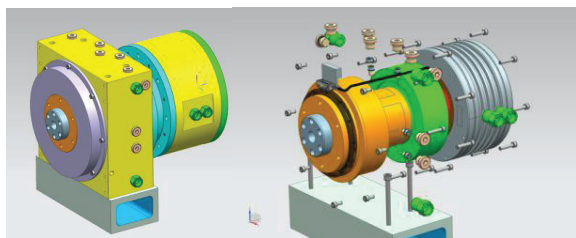


FIGURE 5 CAD images of the Aluminium based R2R hydrostatic bearing units

Hydrostatic bearing design was supported through FEA modelling. This modeling work evaluated different pocket designs and the extent of component distortion at differing pressures and pocket designs. Component distortion was evaluated to ensure the aluminium based design performed with the relatively low Young's modulus of the main spindle components.

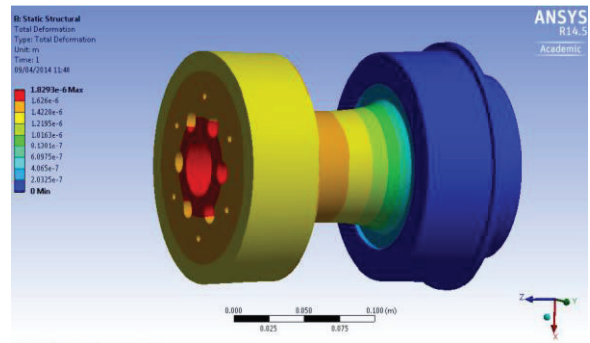


FIGURE 6 FEA modeling of R2R bearing designs (G. Zhou MSc) [7]

Roll embossing – programmable curing

An air bearing “nip” roller control unit (design not yet disclosed) is employed to provide accurate thickness control between embossing and nip rollers.

After embossing of the slot die applied acrylics (and variants) a low temperature curing unit is employed. To minimize necessary heat inducement during curing of applied acrylics a high specification 365nm LED UV curing source was developed. This unit offers intensity and region area control. This large scale UV unit was produced by UV Integration Limited, Oxfordshire, England, see Figure 5. This high specification light source offers up to 9w/cm2 over a programmable 1.41m by 0.046m light zone.



FIGURE 7 UV Integration LED curing unit mounted under embossing roll

The 14kW UV light source is cooled using a 3 stage 20kW capacity (~ 60 litres per minute) temperature control system developed by 3D Evolution of Bedfordshire, England. This large capacity cooling unit offers temperature control

of the UV cure system and other heat generating / absorbing systems of the R2R platform.

Slot die deposition technology

A slot die system is used to apply a range of fluids at 0.5 – 100 micrometre thickness range. Fluids ranging 50 – 3000 mPas viscosity can be applied at flow rates of 3 – 1000ml/minute. The system shown in Figure 6 was specified through film processing research performed by Ixscient Limited and Microsharp Innovations Limited, and manufactured by TSE/Troller of Switzerland.

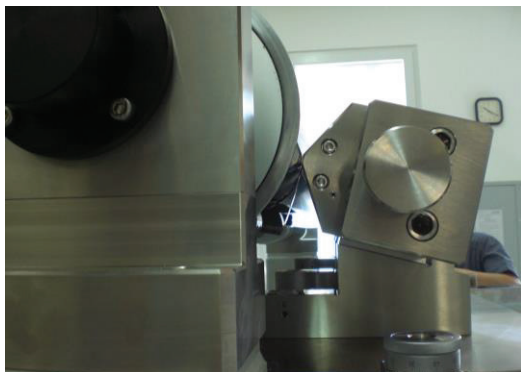


FIGURE 8 Slot die application system

Control of the overall R2R research platform requires a comprehensive and integrated system. The UV curing system, motional control of the web, ink jet, laser and the thermal control of the main elements of the machine are incorporated within a single CNC system. Film positioning will be dealt with through a number of real and virtual axis within the control system.



FIGURE 9 Control units for, web motion, slot die, UV curing intensity, temperature, ink jet printing and laser processing.

CONCLUSIONS

This paper has briefly introduced progress in the design and development of a large scale R2R

research platform. The research platform has been created to provide a basis for experimental testing and optimizing of new precision production methods for the fabrication of ever finer features on a diverse range of films.

The R2R platform has also been defined to provide a large scale passive film fabrication facility. Its ultra precision characteristics enable fine (sharp edged) micro-structured films to be produced. This passive film production process will employ ultra precision control of slot die, roll embossing and controlled UV curing processes.

Creation of the Cranfield R2R research platform has encouraged development of a new manufacturing supply chain. In so doing, awareness of emergent film based products has evolved amongst leading precision engineering companies.

REFERENCES

- [1] Shore P, Morantz P. Ultra Precision “enabling our future. Transcripts of the Royal Society, Phil. Trans. R. Soc. A. 2012 370 3993-4014
- [2] Shore P, et al. Precision engineering of astronomy and gravity research. Journal of Manufacturing technology, CIRP Annals, 59/2/2010, 694-716
- [3] Allsop J, Mateboer A, Shore P. Optimising efficiency in diamond turned Fresnel mould masters. SPIE, Volume 8065, 2011, 806509-11
- [4] Shore P, et al. EPSRC Integrated Knowledge Centre in Ultra Precision and Structured final report, 9/9/2015 at <http://www.ultraprecision.org/wp-content/uploads/2013/12/IKC-reportA4-FINAL-for-website.pdf>
- [5] Shore P, Morantz P, O'Neill W. EPSRC Centre for Innovative Manufacturing in Ultra Precision Mid Term report, 9/9/2015 at <http://www.ultraprecision.org/wp-content/uploads/2014/06/EPSRC-Centre-in-UP-Mid-Term-Report.pdf1>
- [6] Shore P, Morantz P, O'Neill W. EPSRC Centre for Innovative Manufacturing in Ultra Precision 3rd year report, 9/9/2015 at <http://www.ultraprecision.org/wp-content/uploads/2014/06/EPSRC-Centre-in-UP-Mid-Term-Report.pdf1>
- [7] Zhou G, Advanced bearing system for ultra precision plastic electronics production systems, Cranfield University MSc thesis 2014

EQUIPMENT FOR ATMOSPHERIC ATOMIC LAYER DEPOSITION IN ROLL-TO-ROLL PROCESSES

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INTRODUCTION

Amongst thin-film deposition techniques, Atomic Layer Deposition (ALD) has unique properties like high conformality, high layer quality and thickness control down to Å level. Deposition rate, however, is very low in conventional ALD reactors. To achieve high throughput and to reduce costs, there have been recent developments regarding spatial ALD. Whereas in conventional ALD, precursors are dosed separated in time using a purge or pump step, in spatial ALD, precursors are dosed simultaneously and continuously at different physical locations. As purging is no longer needed, the spatial ALD process can be operated at much higher speeds, limited by layer deposition chemistry rather than pumping times. Thus, deposition rates exceeding 1 nm/s have been reported for spatial atmospheric ALD of Al_2O_3 [1]. This has led to the launch of high throughput, industrial scale ALD tools for surface passivation of crystalline silicon solar cells.

Usually, substrates in existing ALD applications are flat and rigid, like silicon wafers or glass plates. Performing spatial ALD on these substrates involves e.g. substrate reciprocation or rotation under a flat ALD injector head. Precursors are not allowed to come into contact with each other, other than on the substrate surface. Separation of precursors is achieved using gas bearing technology, which also enables small precursor chamber volumes.

Because of the increased throughput and decreased cost levels, new application fields are opening up for spatial ALD, such as flexible electronics, including system-in-foil, flexible displays, OLEDs and solar cells. Examples of layers are transparent oxide (semi)conductors (e.g. ZnO), moisture permeation barriers (e.g. Al_2O_3), and buffer layers in CIGS solar cells (e.g. $\text{Zn}(\text{O},\text{S})$).

A new type of atmospheric, spatial ALD reactor has been developed for deposition on flexible substrates [2]. Instead of a flat ALD injector

head, a rotating drum is used to supply the precursor gases to slots at the peripheral surface of the drum, parallel to its rotation axis (see Fig. 1). The foil substrate is transported around the drum surface, where gas bearings are used to separate the foil from the drum as well as separate the different precursors. The foil being contactless enables the drum to rotate at high speed while the foil is slowly advancing. Thus every part of the foil surface will come into contact with a predefined number of precursor cycles. Each individual precursor pair cycle will deposit one monolayer of e.g. Al_2O_3 . In the current design, the drum has six precursor slot pairs at its outer surface. Thus, when the drum rotates at a frequency of 1 Hz, the number of precursor pairs per second is approximately 6, for low foil traversing speeds. When the foil effectively covers ~50% of the drum surface, a ~20 nm thick layer of alumina can be applied in a continuous roll-to-roll process. Such layers can e.g. be used as moisture barrier.

A prototype roll-to-roll ALD reactor has been realized according to the 3D representation in Fig. 2. In this prototype, contactless deposition of alumina has been shown on, amongst others, PET substrates.

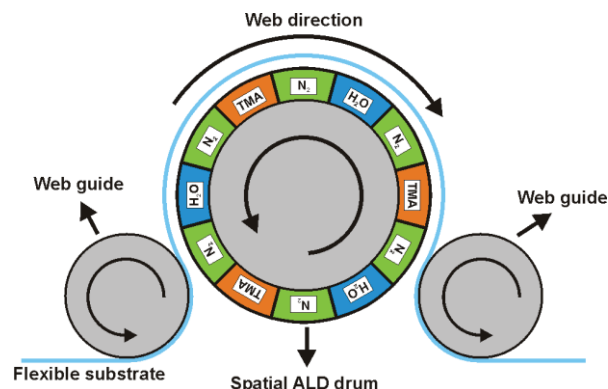


FIGURE 1. Foil moving clockwise over a drum rotating counterclockwise

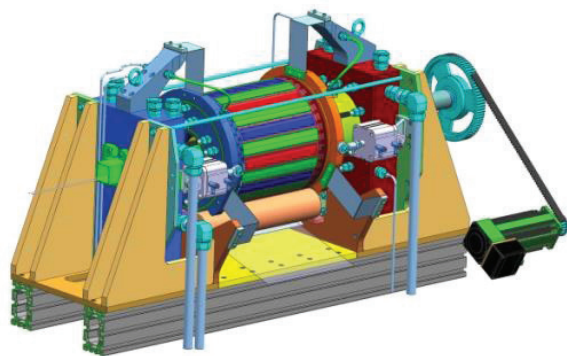


FIGURE 2. 3D representation of a spatial ALD roll-to-roll reactor

ROLL-TO-ROLL ALD FOR CIGS BUFFER LAYERS

As part of the EU FP7 project “R2R CIGS”, a second generation tool is now realized, aimed at deposition of zinc oxysulfide (Zn(O,S)) buffer layers in CIGS solar cells as replacement for CdS (Fig. 3). The foreseen precursors are diethylzinc (DEZn), H_2O and H_2S . The atomic layer deposition of Zn(O,S) using premixing of precursors is enabled by spatial processing [3]. The oxygen to sulfur ratio can be manipulated using variation of precursor premixing ratio in the gas supply as well as variation in exposure time by changing drum rotation speed. Using only DEZn and H_2O , intrinsic zinc oxide (i:ZnO) can be deposited, and by admixing H_2S to H_2O it is possible to deposit a Zn(O,S) compound, which is required for the CIGS buffer layer.

In the ALD reactor, all active chemicals are used in gas phase. The gas supply system for the active gases has supply lines for DEZn, H_2O and H_2S . DEZn is evaporated in a bubbler system using nitrogen and further diluted depending on required process settings. Optionally, the bubbler can be heated to increase evaporation rate. The H_2S supply uses prediluted H_2S in nitrogen as source and is further diluted as required for the deposition process. H_2O is evaporated using a controlled evaporator mixer (CEM) and further diluted in nitrogen. The diluted H_2O and H_2S streams can be mixed to a predetermined ratio for deposition of Zn(O,S) .

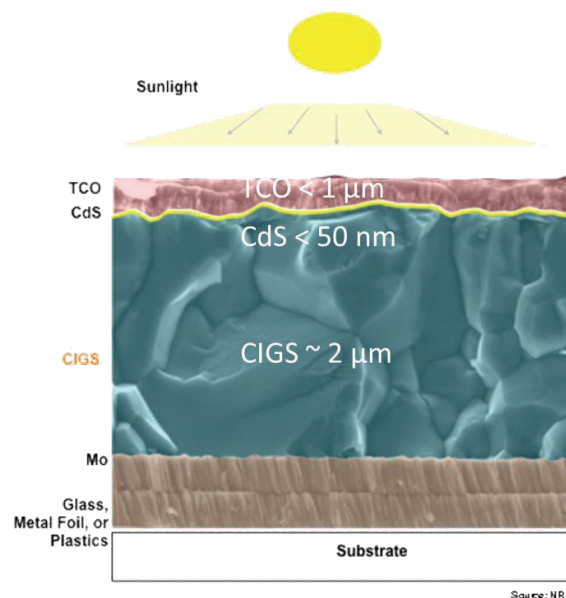


FIGURE 3. CIGS solar cell stack with CdS buffer layer

The ALD reactor output contains reaction products and unused precursor gases, which have to be neutralized. All output gases are fed through a caustic soda scrubber (NaOH solution in water) which neutralizes, both, DEZn and H_2S .

Deposition will be done on polyimide substrates with thickness between 10 and 50 μm at temperatures between 100 and 150 $^\circ\text{C}$. Typical layer thickness to be deposited is between 10 and 80 nm. Other thicknesses are also possible.

WEB HANDLING

The ALD reactor is integrated with a web handling system that enables controlled web speed of 0.1 m/min up to at least 10 m/min, web tension of 10 N and higher, and lateral steering to ensure proper web alignment to the ALD drum. There is no mechanical contact at the deposition side of the web to avoid damage to the active layers. To further improve yield, the machine can be placed in a cleanroom environment.

Fig. 4 shows a schematic of the web handling system. The ALD reactor is schematically shown in the center, inside a double enclosure. The web unwinder, at the left side, is controlling the web speed whereas the rewinder, at the right side, is controlling the web tension. Both winders have an option to apply or remove a protective interleaf (ILF).

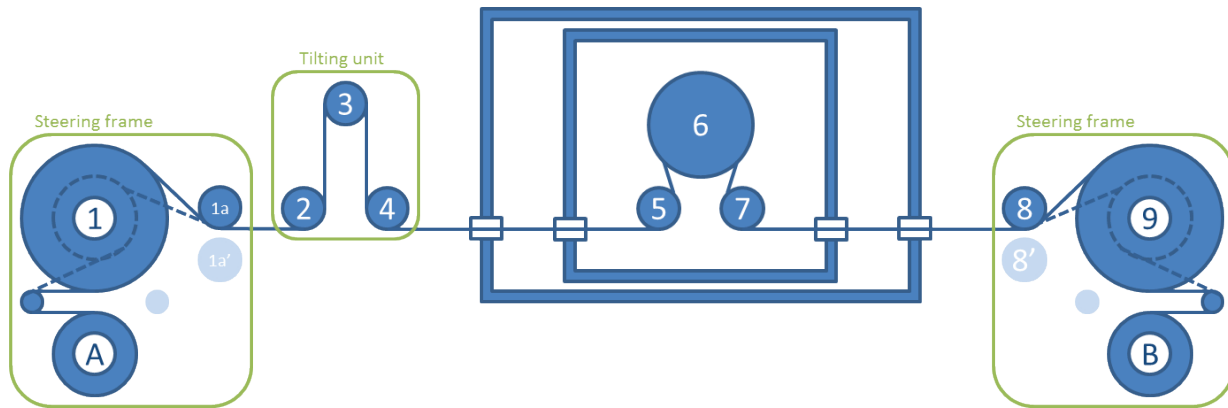


FIGURE 4. R2R ALD web handling system

ENCLOSURES FOR H₂S SAFETY

In normal ALD operation there is no leak of active gases. But because H₂S is poisonous, any accidental leak towards the environment shall be avoided. Therefore, there is a double enclosure around the ALD drum as shown in Fig. 5. Both enclosures have an adjustable exhaust for controlled internal pressure. The pressure in the outer enclosure is just below environment pressure, and the pressure in the inner enclosure is just below outer enclosure pressure. It is not a vacuum system, i.e. pressure levels are approximately atmospheric.

There is an option to purge the enclosure volumes with a clean/inert gas, e.g. nitrogen.

MACHINE CONTROL

The machine is controlled from a central control system, which controls the sub-systems like motion control (web handling, drum motion), heating control (gas heating, component heating), pneumatic control, safety and gas supply & abatement.

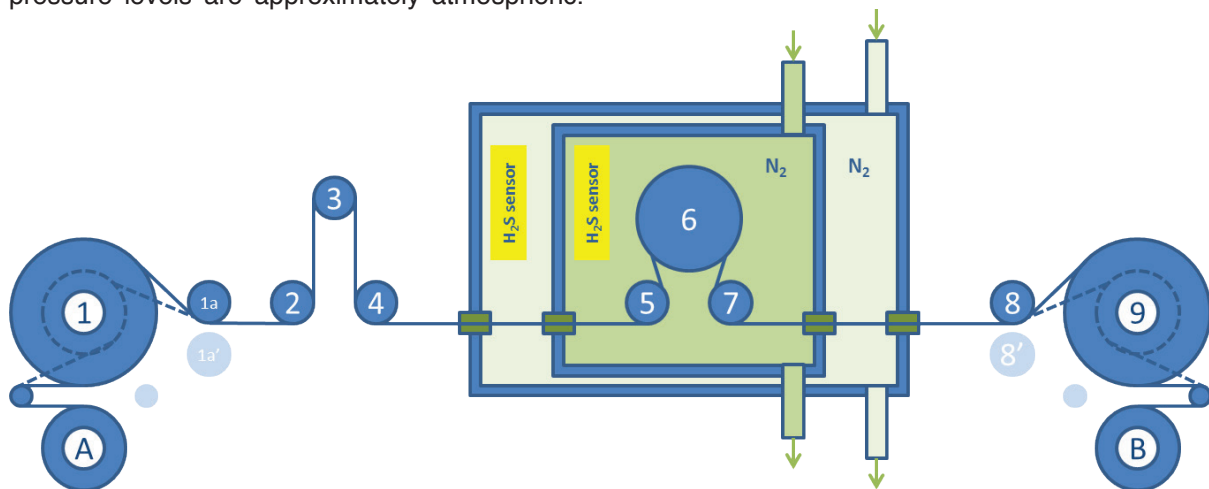


FIGURE 5. Double enclosure for H₂S safety

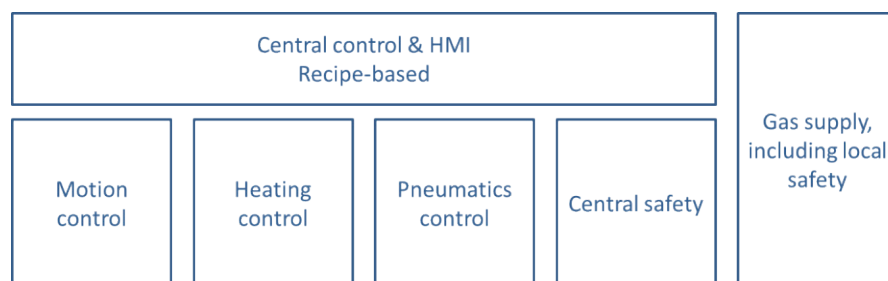


FIGURE 6. Control architecture

CONCLUSIONS AND OUTLOOK

Based on the spatial ALD reactor concept as published in [2], a second generation roll-to-roll spatial ALD reactor has been designed and built for deposition of zinc oxysulfide (Zn(O,S)) buffer layers in CIGS solar cells as replacement for CdS. This development is part of the EU FP7 project "R2R CIGS". The tool incorporates spatial atomic layer deposition using premixed precursors for deposition of mixed compounds. Web handling has been integrated with the ALD reactor to enable web transport with controlled speed, tension and lateral steering. The ALD reactor is furthermore integrated with central control, gas supply and abatement, heating, pneumatics and safety system. First deposition results are expected in the coming months.

Beside CIGS buffer layers, the tool can also be used for other applications such as barrier layers. For deposition of alumina, the zinc precursor could e.g. be exchanged for an aluminum precursor such as trimethyl aluminium (TMAI). A 3D overview drawing is shown in Fig. 7.

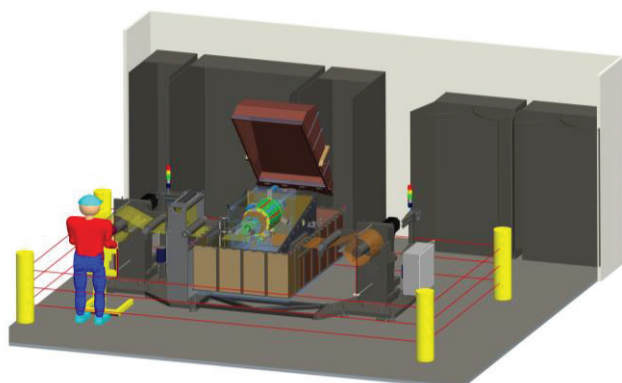


FIGURE 7. Roll-to-roll spatial ALD reactor layout

REFERENCES

- [1] Poodt P, Lankhorst A, Roozeboom F, Spee K, Maas D, Vermeer A. High-speed spatial atomic-layer deposition of aluminium oxide layers for solar cell passivation. *Advanced Materials*, 2010, 22(32), 3564-3567.
- [2] Poodt P, Knaapen R, Illiberi A, Roozeboom F, Van Asten A. Low temperature and roll-to-roll spatial Atomic Layer Deposition for flexible electronics. *Journal of Vacuum Science and Technology, A: Vacuum, Surfaces, and Films*, Vol. 30, 2012, No. 1, p. 01A142-1/5.
- [3] Illiberi A, Scherpenborg R, Wu Y, Roozeboom F, Poodt P. Spatial atmospheric atomic layer deposition of $\text{Al}_x\text{Zn}_{1-x}\text{O}$. *ACS Applied Materials & Interfaces*, 2013, 5(24), 13124-13128.

ROLL-TO-ROLL GRAPHENE TRANSFER: MACHINE DESIGN AND POTENTIAL CHALLENGES

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INTRODUCTION

Since the discovery of the 2D carbon material by its isolation from bulk graphite, graphene has been envisioned to replace indium tin oxide (ITO) to become the next generation transparent electrode for flexible displays due to its exceptional optical transparency, mechanical strength and electrical conductivity [1-3]. Its high carrier mobility has also sparked the research of graphene-based thin-film transceivers [4]. To scale up graphene production, chemical vapor deposition has been widely used to reliably synthesize large-area graphene of high-quality graphene on copper foil under $\sim 1000^{\circ}\text{C}$ [5].

To realize a large-scale manufacturing of graphene-based flexible electronics, the next crucial step is to develop a reliable method for transferring graphene from its growth substrate, copper foil, to target substrates such as PEN and polyimide [6, 7].

In recent years, roll-to-roll (R2R) manufacturing has emerged to be an efficient and scalable solution in the electronics industry to fabricate solar panels, displays, and battery electrodes with high throughput [8-10]. The idea of using R2R manufacturing to mass-produce graphene has also been investigated, and two major designs were reported in 2010 and 2013 [11, 12]. In both designs, the seed copper layer was removed by wet chemical etching. Thermal release tape (TRT) and epoxy were respectively

used as the polymer backing layer during wet etching.

MOTIVATION

Although previously reported R2R systems could effectively transfer large-scale graphene, there are major drawbacks associated with their transfer processes. First, wet etching of high-quality copper foil significantly increases the cost of graphene production. The dissolved copper foil loses its original shape and purity. Second, copper etching is a time-consuming process, and it could cause undesirable doping of graphene. Furthermore, the polymer support used for graphene protection could lead to contamination and/or damage to the transferred graphene. Specifically, TRT could leave organic residue on graphene surface, and graphene could not be separated from epoxy after transfer [13].

Therefore, an efficient non-etching R2R graphene transfer method is necessary to realize low-cost graphene production in large quantity that is suitable for large-scale manufacturing of graphene-based electronics.

NON-ETCHING R2R GRAPHENE TRANSFER

Recently, a novel R2R transfer process was designed to provide solutions to the aforementioned issues with the existing R2R graphene transfer designs.

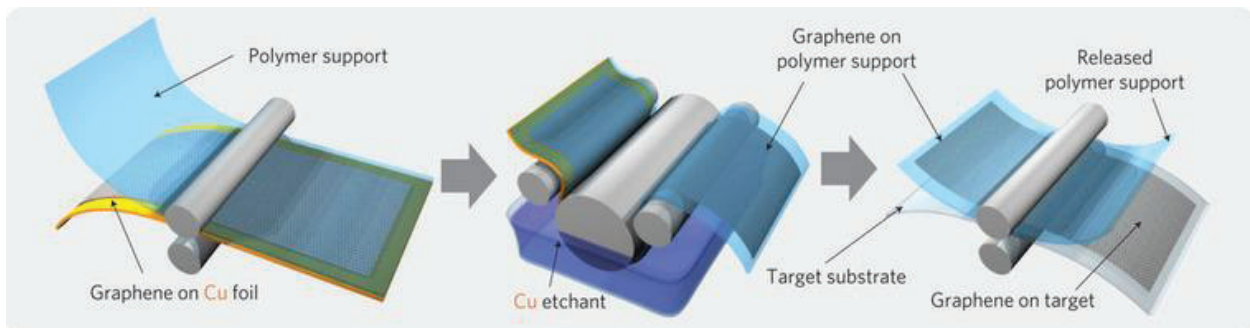


FIGURE 1. Roll-to-Roll graphene transfer with thermal release tape and chemical etching [11].

In this design, wet chemical etching is replaced by electrochemical delamination or dry peeling to preserve copper foil and achieve faster transfer speed [14].

In electrochemical delamination, graphene film on copper foil goes through an electrolytic solution, and a DC voltage is applied to the copper substrate to make it the cathode in water electrolysis, and an inert platinum mesh is used as the anode. When DC power is applied, hydrogen bubbles will form at the interface between copper foil and as-grown graphene, which help to gently push graphene off the copper surface while keeping the copper foil intact for recycling [15]. FIGURE 2 shows the hydrogen bubbles forming on the surface of copper foil in the R2R graphene transfer prototype machine.

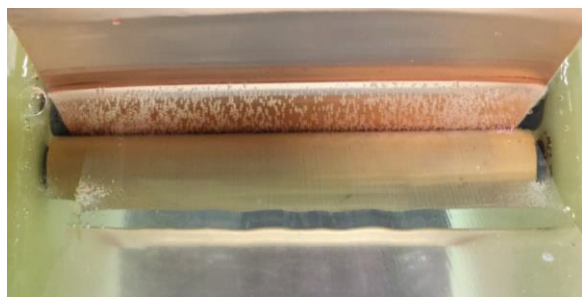


FIGURE 2. Hydrogen bubbles forming on copper foil cathode submerged in electrolytic solution.

After electrochemical delamination, copper foil is recycled for future graphene growth, and the delaminated graphene film on polymer support is rinsed with DI water and then dried. To protect the transferred graphene on polymer backing layer, an additional protection film is added on top of the graphene layer to prevent graphene from touching the back side of the polymer film when it is rolled up on the final roller. FIGURE 3 illustrates the R2R graphene transfer process with electrochemical delamination.

In addition to electrochemical delamination, dry peeling can also be used to transfer graphene directly from copper, and the first R2R graphene transfer prototype machine is also capable of dry peeling transfer. A polymer that has strong bonding effect with graphene, like epoxy, should be used to provide an adequate adhesion to facilitate graphene delamination from its growth substrate. After the dry peeling transfer, copper foil can also be reused to grow graphene.

Both electrochemical delamination and dry transfer can achieve a faster transfer speed than copper etching, so they are more suitable for scaling up graphene production to realize mass production of large-area graphene in the future.

STRAIN-ISOLATION POLYMER SUPPORT

To protect graphene during R2R transfer with electrochemical delamination, a double-layer polymer support composed of a soft elastomer layer and a hard flexible plastic film is used.

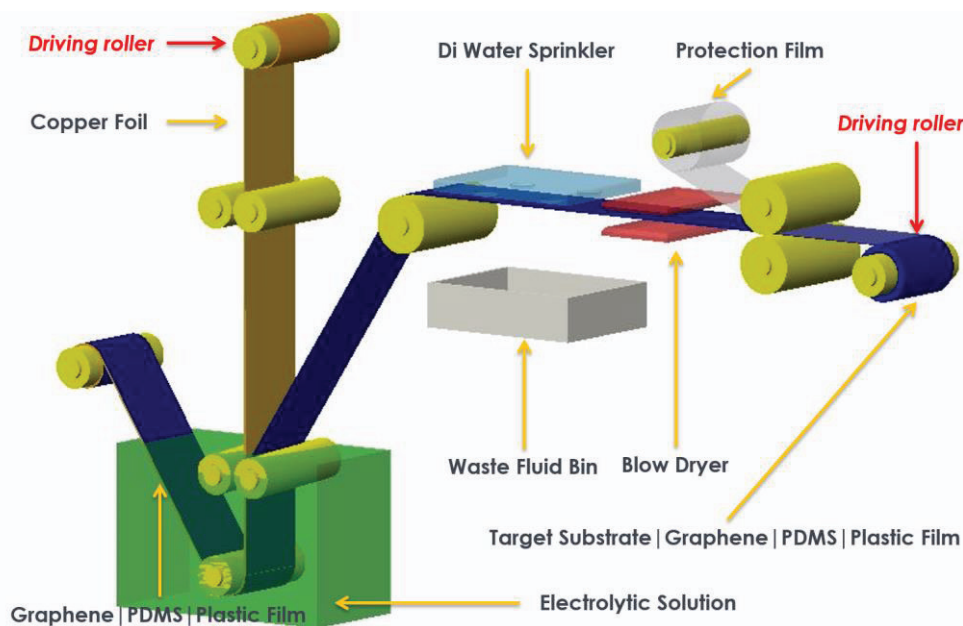


FIGURE 3. R2R graphene transfer with electrochemical delamination.

To prepare the double-layer polymer support, graphene on copper is coated with a liquid PDMS layer with 10:1 curing agent ratio. Compared with solid PDMS film, liquid PDMS precursor can fully conform to its uneven surface generated during high-temperature CVD growth of graphene film.

Next, a PET film is added on top of the coated liquid PDMS. Finally, the sandwich structure is cured under 70°C for 4 hours. FIGURE 4 shows the double-layer polymer support on both sides of the copper foil.



FIGURE 4. Double-layer polymer support used in R2R graphene transfer.

Compared with single-layer polymer support like PMMA or polyimide, the double-layer polymer support could provide a strain-isolation effect that would effectively reduce the tension-induced stress in graphene during R2R transfer [16].

After graphene separates from copper foil by electrochemical delamination, the tension exerted on this sandwich structure will be mostly sustained by the outer plastic film since it has a relative large elastic modulus. Assuming no slipping happens at the graphene-PDMS and PDMS-plastic film interfaces, the strain developed in the plastic film layer will be considerably reduced through the elastomeric PDMS. As a result, the stress exerted on the graphene layer will be largely reduced.

CHALLENGES AND FUTURE PLANS

There are many challenges in developing an effective R2R graphene transfer process. One of the major challenges is to precisely control the transfer speed.

As copper foil and delaminated graphene on the polymer support are being collected onto the driving rollers, the linear speed for both films would increase at a different rate if the motor speed is kept constant because the thickness of copper foil is smaller than that of the transferred graphene with its polymer backing layer. It is

important to maintain a constant web speed in order to guarantee a high-quality graphene transfer since both electrochemical delamination and dry peeling are sensitive to the speed of graphene delamination.

This problem could be addressed by implementing a close-loop control of the motor speed with an optical encoder that measures the instantaneous film speed as the control input.

Two encoders will be attached to two passive guiding rollers in the R2R system. One encoder measures the linear speed of the copper foil, and the other one measures the linear speed of the plastic film. The controller will compare the current film speed to its target value and make adjustments accordingly. With two separate micro-controllers, the speed of the copper foil and the speed of the transferred graphene on polymer support will be adjusted individually.

Furthermore, a roller with piezoelectric sensors will be installed in the system to guide transferred graphene on polymer support, so that the tension in the polymer film could be calculated based on the measurements from the pressure sensitive roller.

Web misalignment could lead to quality degradation in transferred graphene, as experienced in other R2R systems [17]. To improve the alignment of the current prototype, convex rollers will be implemented to create a stress gradient pointing to the roller center. It can help to pull the film toward the middle of the roller when misalignment happens during R2R transfer.

FIGURE 5 shows the first prototype of the non-etching R2R graphene transfer tool designed and built at UT-Austin. This prototype is capable of both electrochemical delamination and dry peeling. The speeds of two driving rollers are adjustable to suit different graphene transfer conditions. Upon finalizing the control system in the machine, a parametric study of the graphene transfer results with different transfer conditions, such as delamination speed, current density and film tension, will be performed. A design analysis of this prototype machine will be carried out to help realize the non-etching R2R production of electronic grade graphene sheet in the near future.

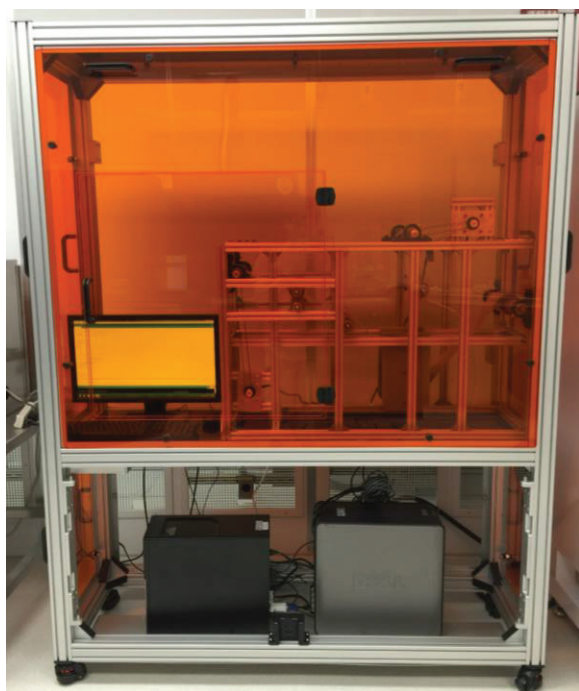


FIGURE 5. First prototype of non-etching R2R graphene transfer machine.

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REFERENCES

- [1] Novoselov, K. S. A., et al. "Two-dimensional gas of massless Dirac fermions in graphene." *nature* 438.7065 (2005): 197-200.
- [2] Geim, Andre K., and Konstantin S. Novoselov. "The rise of graphene." *Nature materials* 6.3 (2007): 183-191.
- [3] Lee, Changgu, et al. "Measurement of the elastic properties and intrinsic strength of monolayer graphene." *science* 321.5887 (2008): 385-388.
- [4] Neto, AH Castro, et al. "The electronic properties of graphene." *Reviews of modern physics* 81.1 (2009): 109.
- [5] Li, Xuesong, et al. "Large-area synthesis of high-quality and uniform graphene films on copper foils." *Science* 324.5932 (2009): 1312-1314.
- [6] Choi, Dukhyun, et al. "Fully rollable transparent nanogenerators based on graphene electrodes." *Advanced Materials* 22.19 (2010): 2187-2192.
- [7] Luong, Nguyen Dang, et al. "Enhanced mechanical and electrical properties of polyimide film by graphene sheets via in situ polymerization." *Polymer* 52.23 (2011): 5237-5242.
- [8] Søndergaard, Roar, et al. "Roll-to-roll fabrication of polymer solar cells." *Materials today* 15.1 (2012): 36-49.
- [9] Liang, R. C. et al. "Microcup® displays: Electronic paper by roll-to-roll manufacturing processes." *Journal of the Society for Information Display* 11.4 (2003): 621-628.
- [10] Chaug, YS al, et al. "Roll-to-Roll Processes for the Manufacturing of Patterned Conductive Electrodes on Flexible Substrates." *MRS Proceedings*. Vol. 814. Cambridge University Press, 2004.
- [11] Bae, Sukang, et al. "Roll-to-roll production of 30-inch graphene films for transparent electrodes." *Nature nanotechnology* 5.8 (2010): 574-578.
- [12] Kobayashi, Toshiyuki, et al. "Production of a 100-m-long high-quality graphene transparent conductive film by roll-to-roll chemical vapor deposition and transfer process." *Applied Physics Letters* 102.2 (2013): 023112.
- [13] Wang, Di-Yan, et al. "Clean-Lifting Transfer of Large-area Residual-Free Graphene Films." *Advanced Materials* 25.32 (2013): 4521-4526.
- [14] Kang, Junmo, et al. "Graphene transfer: key for applications." *Nanoscale* 4.18 (2012): 5527-5537.
- [15] Wang, Yu, et al. "Electrochemical delamination of CVD-grown graphene film: toward the recyclable use of copper catalyst." *ACS Nano* 5.12 (2011): 9927-9933.
- [16] Kim, Dae-Hyeong, et al. "Ultrathin Silicon Circuits With Strain-Isolation Layers and Mesh Layouts for High-Performance Electronics on Fabric, Vinyl, Leather, and Paper." *Advanced Materials* 21.36 (2009): 3703-3707.
- [17] Zhang, Hao, et al. "Micron-Sized Feature Overlay Alignment on Large Flexible Substrates for Electronic and Display Systems." *Display Technology, Journal of* 7.6 (2011): 330-338.

CONTROL STRATEGY FOR FLEXURE-BASED PRECISION ROLL-TO-ROLL MACHINE

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In this paper, we present the control strategy for a flexure-based precision roll-to-roll machine, where microcontact printing is adapted to this platform for fabricating photonic devices. To achieve nanometer level pattern resolution over a large area, it is critical to precisely control the contact time and uniformity of contact force. Accordingly, we implement different control methods, including decoupling control, input shaping, and cascade control, to optimize the system performance, i.e. to minimize settling time and motion coupling without overshoot.

ISSUES WITH PARASITIC MOTION

Figure 1 shows the prototype of the flexure-based roll-to-roll (R2R) machine reported in [1, 2]. The CAD model of the core positioning stage is presented in Figure 2, where two compliant X-Y stages are installed to support the two ends of a print roller with air bearing and integrated load cells. Although each X-Y stage presents a decoupled design, unwanted motion coupling at micron to submicron level is still present due to manufacturing errors and imperfect constraint and loading conditions. In addition, motion coupling across different X-Y stages is present due to a rigid connection of the print roller. The coupling behavior has been investigated and partially resolved with a PID controller [2]. However, the settling time was relatively long and the overall performance can be further improved. In this paper, we address this issue with different control strategies. As the Y motions of the two X-Y stages critically control the uniformity of printing force, we present all results in the Y direction.

ISSUES WITH PID CONTROL

As shown in Figure 3, the conventional control scheme for the positioning stage, i.e. the two X-Y stages, employ two PID controllers, i.e. $C_{11}(s)$ and $C_{22}(s)$. $P_{12}(s)$ and $P_{21}(s)$ describe the interactions between the two stages due to the rigid connection of the print roller. As the plant model does not change during the control process and steady-state errors can be

eliminated by the PID controllers, the coupling effect may be suppressed and the X-Y stages can move to desired positions. However, in this control scheme, a coupled feedback loop [3] is present, i.e. $r_1 \rightarrow y_2 \rightarrow r_2 \rightarrow y_1 \rightarrow r_1$. This can cause unwanted coupling motion, prolonged settling time, and sometimes instability.

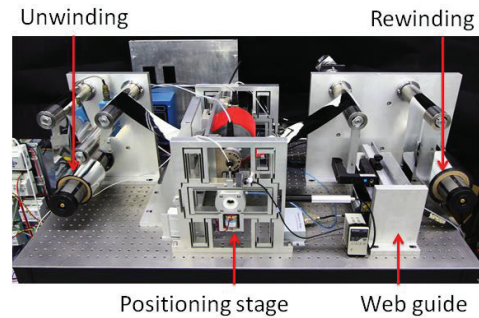


FIGURE 1. Flexure-based roll-to-roll system

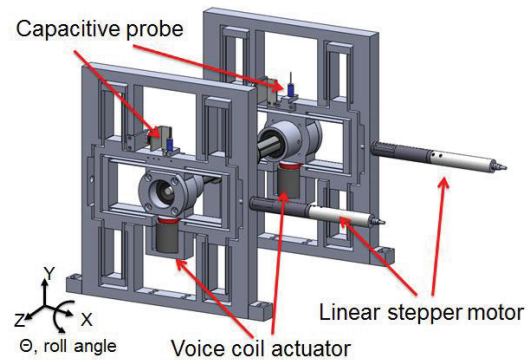


FIGURE 2. CAD model of the positioning stage with integrated actuators and sensors

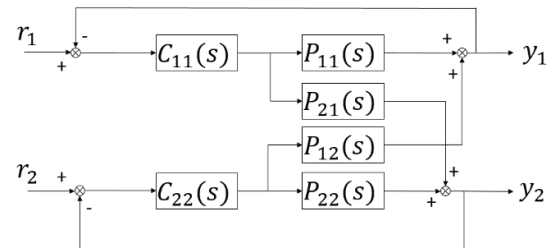


FIGURE 3. Block diagram of the positioning stage with PID control

DECOUPLING CONTROL

To improve the performance of our R2R system, decoupling control [3] is used to minimize the settling time. Figure 4 shows the block diagram of the decoupling control, where two decouplers, i.e. $D_{21}(s)$ and $D_{12}(s)$, are employed in the system to compensate the coupling motion. The transfer function of the decouplers can be expressed as

$$D_{21}(s) = P_{21}(s)/P_{22}(s) \quad (1)$$

$$D_{12}(s) = P_{12}(s)/P_{11}(s) \quad (2)$$

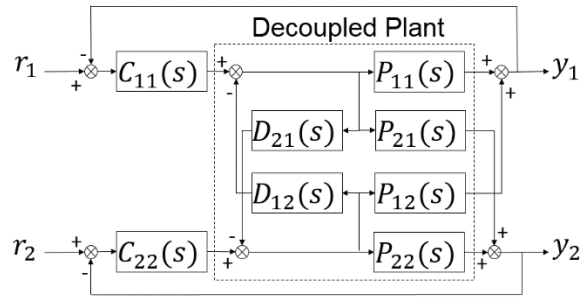


FIGURE 4. Block diagram of the positioning stage with decoupling control

The accuracy of the decouplers relies on the accuracy of the compliant stage and coupling model. To obtain precise models, frequency response of the compliant stages and coupling behavior are characterized experimentally. Next, Fourier analyses are performed and the results are plotted in Figure 5 along with fitted frequency response. In Figure 5, we find the fitted frequency response of the model matches the experimental data well in low frequency region, although distortions occur in high frequency region, which is caused by nonlinearities of the system, e.g., electronic noises, nonlinearities of the actuators and drivers.

As a steady contact force is desired during the actual microcontact printing process, the X-Y stages work in the low frequency range. Accordingly, the distortion between experimental results and fitted models in high frequency region can be neglected. The decouplers can thus be designed as the ratio of the gains of the transfer functions in the low frequency region, e.g. $D_{21}(s) = P_{21}(s)/P_{22}(s)$.

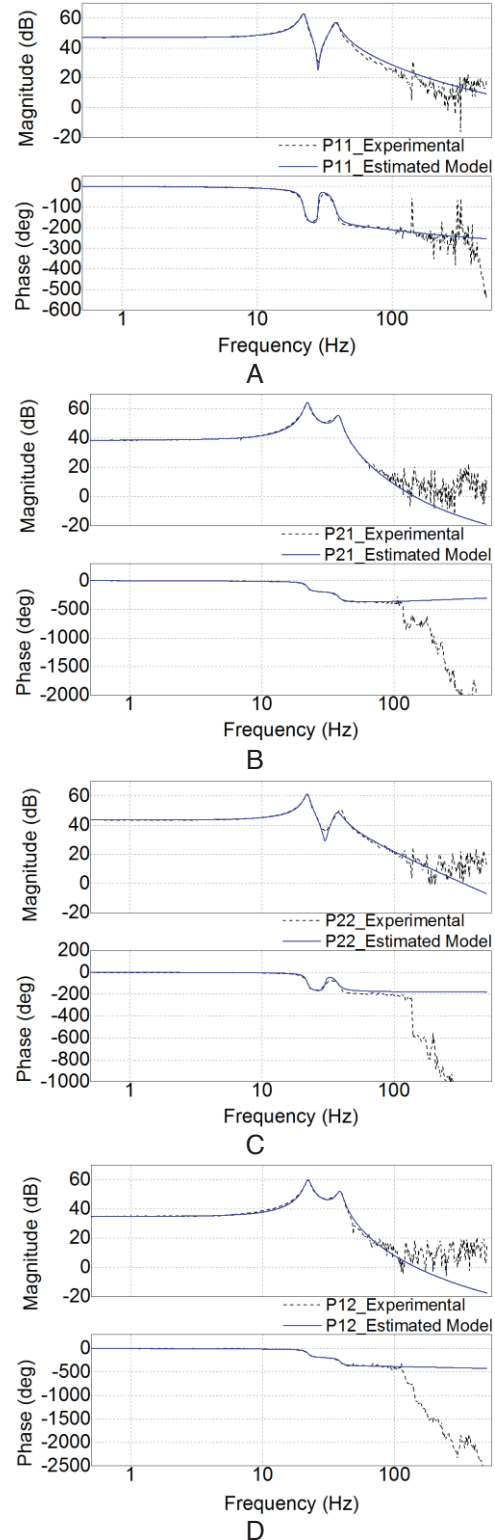


FIGURE 5. Measured frequency response (dotted line) and fitted transfer function (blue line) of $P_{11}(s)$, $P_{21}(s)$, $P_{22}(s)$ and $P_{12}(s)$ shown in A, B, C, D respectively.

Figure 6 presents the experimental results comparing the PID and decoupling control. In the experiment, X-Y stage 1 is commanded to move in the Y direction in 10 μm steps, while X-Y stage 2 remains at the same position. In Figure 6A, we find the conventional PID controller cannot effectively remove the coupling behavior, and a 1.59 μm motion is coupled to X-Y stage 2 (15.9% motion coupling). (Note without the PID controller, the motion coupling is around 34% [2].) In Figure 6B, when decoupling control is employed, we find the coupling motion can be controlled within 100 nm (1% motion coupling).

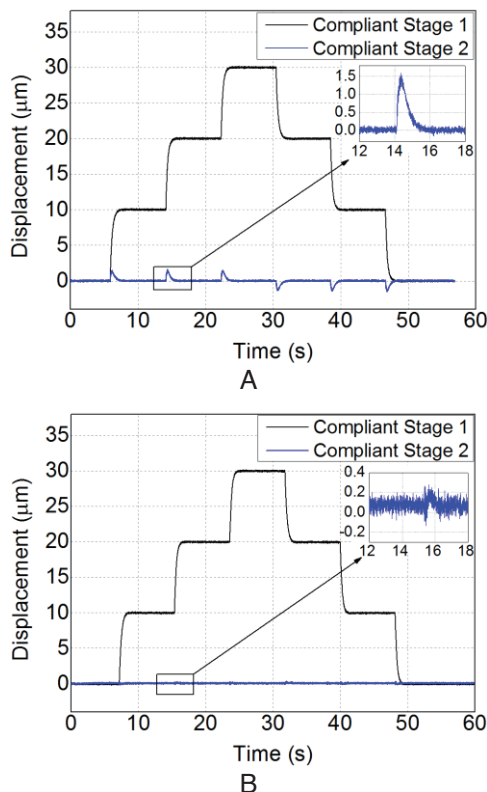


FIGURE 6. Motion decoupling experiments using the PID control (A) and decoupling control (B)

INPUT SHAPING

Input shaping is a variation of the feed forward control method which is used to reduce or eliminate endpoint vibration by not exciting the natural frequencies [4]. This method can be applied to both open and closed loop control for arbitrary set-point. To demonstrate the effectiveness of vibration and overshoot suppression with an input shaping controller, we compare the experimental results of open loop control, closed loop (PID) control and input shaping on a single X-Y stage. Figure 7A

presents the experimental results of a step response with three control methods respectively. With open loop control, we observe a more than 25 μm overshoot with ~ 1 second settling time. With closed loop control, the overshoot is eliminated and the settling time remains unchanged. With input shaping, we find the overshoot is entirely eliminated and the settling time is reduced to within 0.1 second. By using Fast Fourier Transform (FFT) to convert the time domain information to frequency domain, the vibration amplitudes on different frequencies can be shown directly. Figure 7B presents the frequency spectrum of the step response. Although the closed loop control is able to suppress the vibration near the natural frequency effectively, we find input shaping achieves better performance because the frequency spectrum is smoothed distinctly and the vibration is largely suppressed near the natural frequency.

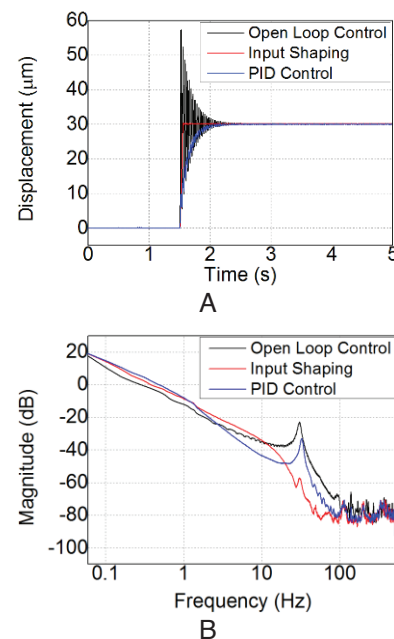


FIGURE 7. Input shaping control experiments. A: Step response with open loop control, input shaping and PID control. B: Frequency spectrum with open loop control, input shaping and PID control

Next, we combine the PID and input shaping control in hope for better results. As shown in Figure 8, there are two possible controller configurations, i.e. placing the input shaping filter outside (position #1) or inside (position #2) the loop, referred as “outside-the-loop input shaping” (OLIS) and “closed-loop signal shaping” (CLSS) respectively [5]. Note that without the input

shaping filter, Figure 8 forms a classic PID control scheme.

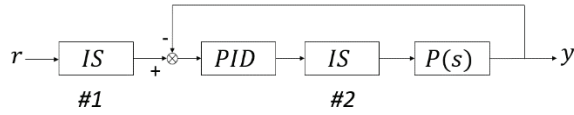


FIGURE 8. PID control combined with input shaping filter of two possible configurations: (1) outside-the-loop input shaping (OLIS) and (2) closed-loop signal shaping (CLSS)

Figure 9 presents the experimental results of a step response using (1) PID, (2) OLIS, and (3) CLSS control. It is found in Figure 9A that the CLSS control results in a longer settling time compared to other methods. This is because the input shaping filter inside the loop prolongs the processing time which cannot be solved by adjusting the PID parameters. In contrast, the OLIS achieves a better performance than the PID control, where the vibration along the rising curve is effectively suppressed with slightly compromised settling time. Figure 9B shows the frequency spectrum of the step response, where both the CLSS and OLIS control effectively reduce the vibration magnitude near the natural frequencies. Therefore, OLIS control is a better choice over PID and CLSS control for our system.

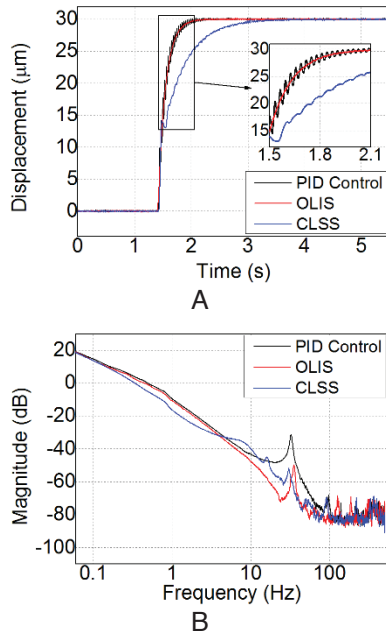


FIGURE 9. Input shaping and PID control. A: Step response with PID, OLIS and CLSS control. B: Frequency spectrum with PID, OLIS and CLSS control

CASCADE CONTROL

In this section, we combine the decoupling, PID, and input shaping control in cascade control to further minimize the system response time and cross-stage motion coupling and use it in a practical printing process. As shown in Figure 10, two control loops, i.e. the master loop and the slave loop, are applied to control the contact force and roller position respectively. The desired contact force is first set in the master loop which then generates a reference position set-point for the slave loop.

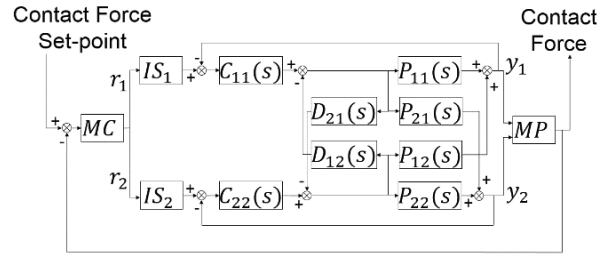


FIGURE 10. Block diagram of the cascade control. MP: Master process; MC: Master controller; IS: Input shaping filter

Figure 11 presents the experimental results of the cascade control. In this experiment, the two X-Y stages are programmed to generate (1) uniform force steps of 500 mN intervals, and (2) displacements of 10 μm intervals; uniform forces and displacements are considered achieved when the print roller is parallel to the impression roller. Before the experiment, all the position sensors, load cells, the roller positions have been carefully calibrated to ensure good parallelism. Capability to generate precise and uniform forces are important as different patterning resolution and speed require different printing forces. From the inset in of Figure 11A, it can be observed that uniform forces have been achieved and controlled within ± 0.01 N precision. Figure 11B presents the displacement steps. From the inset in of Figure 11B, it can be observed that uniform displacements have been achieved and controlled within ± 100 nm precision.

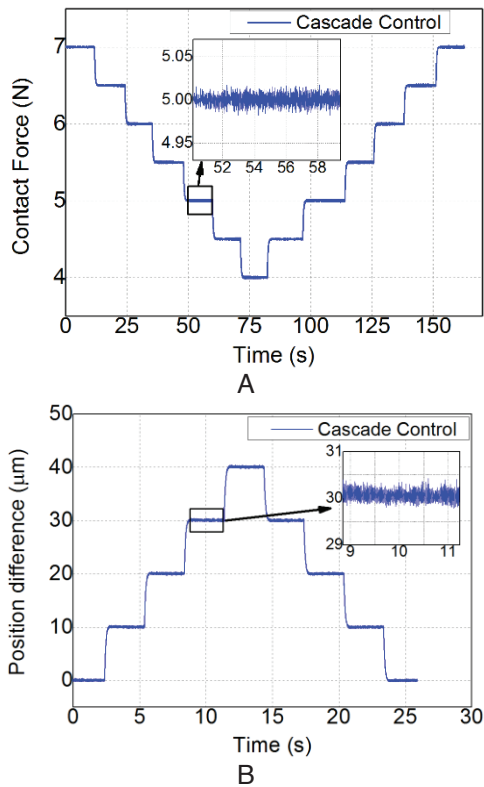


FIGURE 11. Experimental results of the cascade control. A: Uniform force steps of 0.5 N intervals. B: Displacement steps of 10 μm intervals

CONCLUSION

In this paper, we have implemented different control strategies, including decoupling control, input shaping, PID, and cascade control, to improve the dynamic performance of the flexure-based roll-to-roll system. The experimental results have shown that the combination of the decoupling control, PID, input shaping as well as the cascade control can effectively eliminate the cross-stage motion coupling to within 1% and substantially improve the precision of the force and displacement control versus conventional PID controllers.

ACKNOWLEDGMENT

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REFERENCES

- [1] Zhou X, Cheng J, Zhao N, and Chen S. A Flexure-Based High-Throughput Roll-to-Roll Printing System. Proc. of the Annual Meeting of the ASPE, Saint Paul, MN, USA, Oct. 2013, pp.353-57.

- [2] Zhou X, Cheng J, Zhao N, and Chen S. A Flexure-Based Roll-To-Roll Machine for Fabricating Flexible Photonic Devices. Proc. of the Annual Meeting of the ASPE, Boston, MA, USA, Nov. 2014, pp.395-98.
- [3] Seborg D E, Mellichamp D A, Edgar T F, et al. Process Dynamics and Control. John Wiley & Sons. 2010.
- [4] Singer NC, Seering WP. Preshaping Command Inputs to Reduce System Vibration. Journal of Dynamic Systems Measurement and Control. 1990: 112(1): 76-82.
- [5] Huey J R. The intelligent combination of input shaping and PID feedback control[J]. 2006.

PRECISION ENGINEERING OF MULTI-LAYER OPTICAL FILMS

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ABSTRACT

Multiple material and process variables interact and affect the performance of multi-layer optical films manufactured with roll to roll processes. Precision engineering and precise control of these variables has enabled an elite class of mirrors and reflective polarizers. Mathematical modeling of new optical film concepts provides entitlement targets for what light and energy management innovations are possible if all the variables are optimized and precisely controlled. Often, new polymer materials are needed to provide the optical, chemical, and mechanical properties required for different applications. Precise polymer extrusion is critical to minimize particulate defects and to minimize layer thickness variation. Fluid-mechanic mathematical modeling enables design of multi-layer feed blocks necessary to create hundreds of nano-thickness layers having precise layer uniformity. Cast film web handling and film orientation equipment design improvements have been critical to developing films uniform films with unique and uniform optical properties. Both in-line and off-line properties of these optical films are measured, analyzed, and used in feedback control schemes as illustrated in Figure 1.

Continuous improvement in layer thickness uniformity control has provided more efficient use of materials, enabled thinner films to be made for thinner devices, and the ability to lower cost, or use more expensive materials for higher performance mirrors and reflective polarizers.

In this paper, we describe advancements in the precision engineering of multi-layer optical films which have enabled the development of UV(Ultra-Violet) Mirror Films created with hundreds of layers having optical layer thicknesses less than 100 nanometers, and the most reflective broadband solar mirror films in the world.

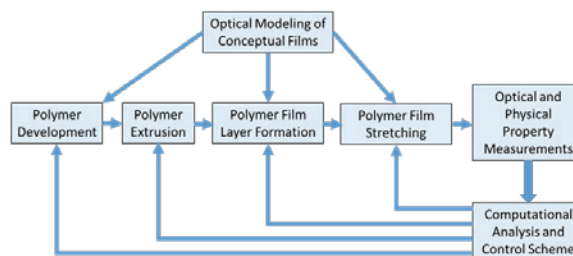


FIGURE 1. Interactive Virtual Model of Precisely Manufactured Multi-Layer Optical Films using a Roll to Roll process..

HIGH LEVEL OVERVIEW ON PRECISION ENGINEERING OF MULTI-LAYER OPTICAL FILMS

Typically, new multi-layer optical film concepts are modeled with constructive interference stack codes capable of accurately predicting entitlement reflectivity that can be achieved and the dependence of that reflectivity on incident light angle. This optical modeling is critical for determining the minimum number of optical polymer layers needed based on the refractive indices of the polymers selected. This process starts with optical modeling and then goes through an iterative loop that matches materials and process to an optical and physical design. One might assume that only polymers having the highest refractive index available and polymers having the lowest available refractive index would be paired for creating the most reflective mirrors. However, properties other than reflective power often dictate the selection of polymers for creating multilayer optical films and thus is the case with UV mirror films and broadband solar mirror films. Other considerations and constraints include but are not limited to UV stability, rheology matching between polymers, adhesion between polymers being coextruded, and thermal stability.

PEN (polyethylene naphthalate) has the highest refractive index of commercially available polymers. PEN also has a UV absorption band edge at 390 nanometers and an absorption tail that significantly absorbs blue light, making it impossible to protect from photo-degradation with commercially available UVAs (ultra-violet light absorbers). Polymers having UV absorption band edges which can be protected from photo-degradation with UVAs have lower refractive indices than PEN. PET (polyethylene terephthalate) has a UV absorption band edge below 350 nanometers and thus can be protected from UV degradation with commercially available UVAs having absorption band edges between 350nm and 400nm without significantly affecting visible light reflectivity. Since the refractive indices of PET are lower than PEN, more optical layers are required to create a visible mirror film with greater than 99% reflectivity, and even more importantly, efficient use of the optical layers. Advances in optical polymer layer thickness control with improved multilayer feedblock and die designs have enabled PET visible mirrors having greater than 99% reflectivity. More efficient use of polymer optical layers coextruded in roll to roll film making processes equates to less total polymer layers and thus less loss of light due to polymer light absorption, and less material also means lower cost products. Examples of more efficient generation of optical layers are shown in Figure 2 and Figure 3.

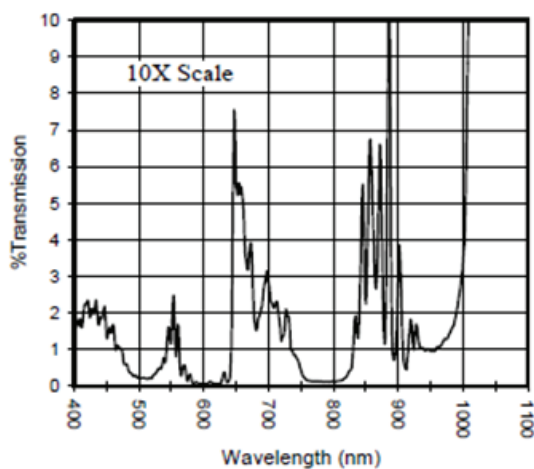


FIGURE 2. Multilayer optical film before improved layer thickness control.

When coated with silver to reflect IR (infrared) light, these PET multilayer optical film visible mirrors are capable of reflecting 96.5% of the total energy available from the sun. Since 4% of the sun's energy is in the UV region, absorption of UV light reduces the total solar reflectivity potential of broadband mirror films made with PET optical layers for reflecting the visible light. Thus there is a need for UV reflecting layers made with inherently UV stable polymers that are less dependent on UVAs for protecting them from UV degradation. Fluoropolymers and acrylate polymers absorb much less UV light and thus are more inherently UV stable. Fluoropolymers require no UV protection and the UV absorption band edge of PMMA (polymethylmethacrylate) is less than 320 nanometers enabling it to be protected from UV degradation with UVAs having absorption band edges between 300 nanometers and 350 nanometers. Multilayer optical film UV mirror films reflecting UV light in the range of 350-400nm placed over PET visible light mirrors enable an increase in total solar reflectivity from 96.5% to 98%.

Based on quarter wave constructive interference physics (1), reflecting shorter wavelengths of light requires thinner optical layers which then requires even more precise control of the optical layer thicknesses. Multilayer optical film UV mirror films made to reflect 350-400nm, as shown in Figure 4 and Figure 5, require optical polymer layer thicknesses in the range of 55-80nm.

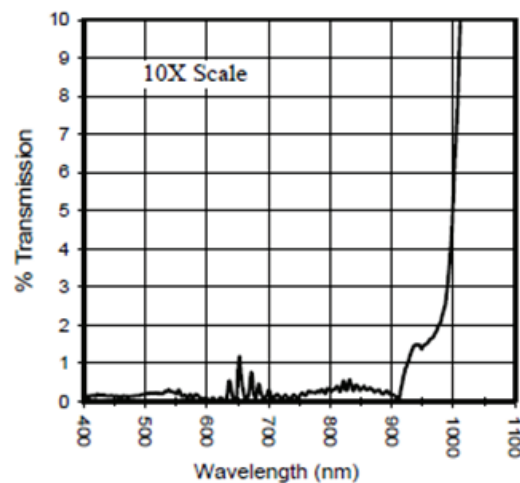


FIGURE 3. Multilayer optical film after improved layer thickness control.

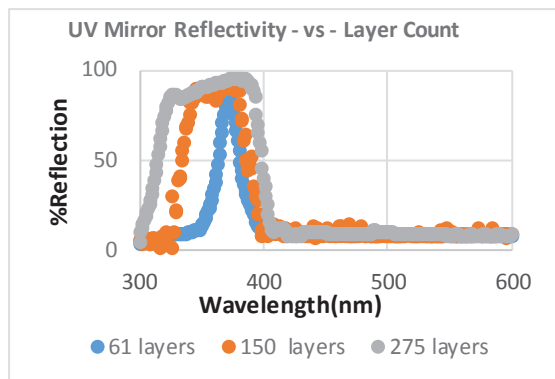


FIGURE 4. Modeled reflectivity of UV Mirror.

Polymer extrusion is at the heart of the multilayer optical film. Extrusion of plastics is commonplace and represents a very large industry. Multilayer optical film manufacturing not only requires expertise in extrusion and materials, but it also requires precision control of layers. Very small deviations from an optical layer design can lead to significant spectral issues. An example of this is in the control of F-ratio. F-ratio is the relative phase thicknesses of the material components in a unit cell. A measure of relative phase thicknesses, is the ratio of the phase thickness for each layer relative to the aggregate phase thickness of the repeating unit cell. It determines how the Fresnel reflections of each layer interface are coherently summed across the unit cells in the optical stack, which in turn determines reflection band behavior with changing incident angle. In many optical stack designs, suppression of higher order reflection bands (harmonics of the primary, first-order reflection band) is an important consideration. As an example, shown below in Figure 6 and Figure 7 is a 400 layer quarter wave design at an F-Ratio of 0.5 as compared to the same at an F-Ratio of 0.515.

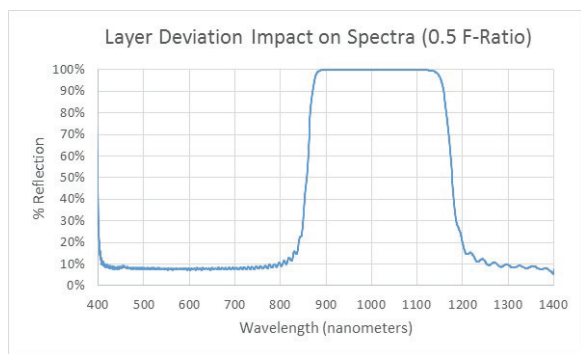


FIGURE 6.

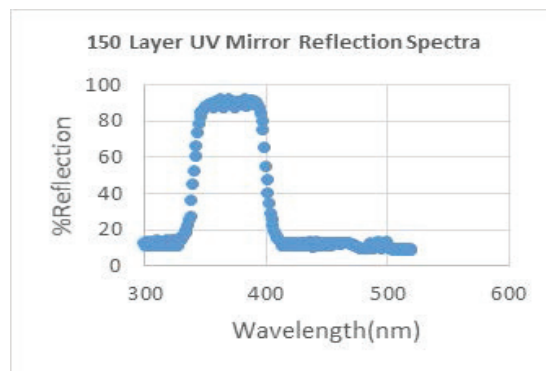


FIGURE 5. Measured reflectivity of UV Mirror.

In the case of Fig. 7, the low index layers are approximately 5 nm thinner than they should be, and the high index layers are approximately 5 nm thicker than they should be. This results in 2nd order reflected color. Instead of this film being colorless, it has significant color. Another example is with a 711 IR reflector. In this case, the “1” layers are as thin as 15 nm. Tolerances are even tighter in this case. Even a few nanometers can make a big difference.

Precision polymer film stretching is a very important element to making a multilayer optical film. Small deviations in stretch ratio in either the machine direction or in the orthogonal in plane direction, will alter the precision extruded layers in an undesirable way. As an example, if the desired stretch ratio was 330% and the actual range of stretch ratio was from 325% to 335%, then one could expect a full 3% variation in layer thickness from this stretching process alone. For some multilayer optical film applications, this may be too much.

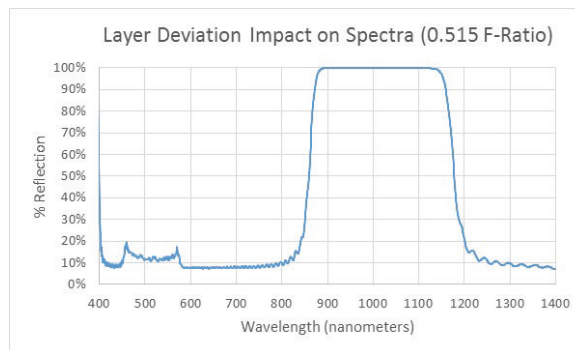


FIGURE 7.

Multilayer optical films have a multitude of product attributes that can be very important. These can range from physical properties like modulus or interlayer adhesion to optical properties like percent transmission over a range of wavelengths for a given source. Precise measurements of these properties is required for equipment design and for process control. In addition to being able to measure in a precise way, it is also important to be able to characterize spatial variability. As mentioned above, very small variations anywhere in the process will lead to variations in the film, and small variations in the film can define whether the film will work for an application or whether it will not.

Precision measurements of certain properties do not necessarily help guide the process. It often requires more than this. A multilayer optical film is being produced that is designed to reflect near IR wavelengths and to transmit visible wavelengths. This film is measured with a precision device which indicates that some visible color exists. Unfortunately, there are many reasons why this color might exist, so additional layer information is required to provide insight on what might need to be done to suppress this color. In other words, some precision measurements simply provide a good indicator whether a film is good or bad, but may not provide much insight as to what is required to fix the problem. In this case, both types of precision measurements may be required in order to manufacture a product like this. Atomic force microscopy (AFM) and transmission electron microscopy (TEM) are two methods that can be used to quantify a layer thickness profile. Shown in Figure 8 is a TEM image from a multilayer optical film.

As mentioned above, precision engineering can enable significant reductions in multilayer optical film thickness. An example of this can be seen with 3M reflective polarizers. The original 3M reflective polarizers for Liquid Crystal Displays were approximately 132 microns in thickness. There are current 3M reflective polarizers that range in thickness from approximately 31 microns all the way down to 17 microns in thickness. This represents a reduction of more than 100 microns, or up to 87% reduction in thickness. Thickness reductions in reflective polarizers have been critical to meet customer demands for thinner and lighter mobile devices.

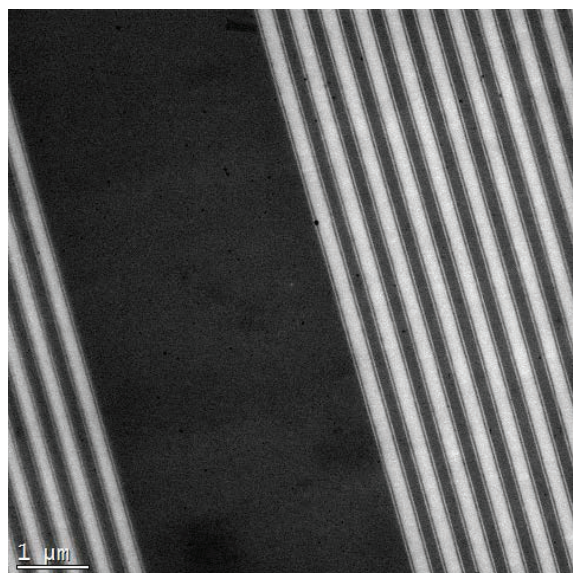


FIGURE 8. This image shows the partial layer structure for a multilayer optical film consisting of both thick and thin layers, all of which are controlled to a tight tolerance. In this case, the two groupings of layers are separated by a much thicker layer.

CONCLUSIONS

Continuous improvement in layer thickness uniformity control has provided more efficient use of materials, enabled thinner films to be made for thinner devices, and the ability to lower cost, or use more expensive materials for higher performance mirrors and reflective polarizers. Advancements in the precision engineering of multi-layer optical films which have enabled the development of UV Mirror Films created with hundreds of layers having optical layer thicknesses <100nm, and one of the most reflective broadband solar mirror films in the world. Not only does precise control of layer thicknesses of coextruded polymers make more efficient use of materials, it enables new to the world products.

REFERENCES

- [1] M.F. Weber, R.J. Strharsky, C.A. Stover, L.R. Gilbert, T.J. Nevitt, A.J. Ouder Kirk, "Giant Birefringent Optics in Multi-layer Optical Films", *Science* **287**, pp 2451-2456, 2000.
- [2] T.J. Nevitt, M.F. Weber, "Recent Advances in Multilayer Polymeric Interference Reflectors", ICCG9 Thin Solid Film Edition, 9th International Conference on Coatings on Glass and Plastics, June 27th 2012.
- [3] T.J. Hebrink, "Durable Films for Improving the Performance of Solar Power Generating Systems", Thrd Generation Photovoltaics, InTech Publisher, March 16th 2012.

Fabrication of Fine Pitch Vee-Groove Patterns on Drum Masters for Roll to Roll BEF Fabrication

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INTRODUCTION

Display technologies are advancing rapidly and color gamut is a key differentiator. Quantum dot enhancement films (QDEF) that significantly extend the color gamut are becoming more economically viable and production of support technologies becomes a critical barrier to entry. One important support system is the brightness enhancement films (BEF). BEF layers increase the brightness of back lights resulting in power savings or thermal management. A typical LCD panel using quantum dots[1] is shown in Figure 1. The BEF film lies between the quantum dot enhancement film and the double brightness enhancement film (DBEF). Typical BEF films used in standard LCD technology have groove pitch spacing of 50 micrometers, but QDEF technology requires very fine pitch spacing of less than 15 micrometers. This reduction in pitch spacing is beyond the reach of commercially available production technologies. This paper addresses the challenges and solutions associated with the fabrication of fine pitch, vee-groove patterns for BEF film production.

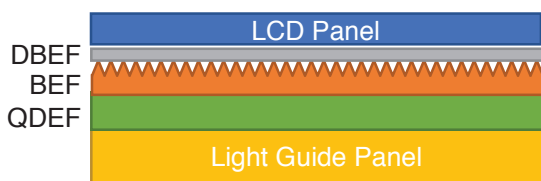


Figure 1. Drawing of a typical structure for quantum dot LCD displays

For high production demands BEF films are manufactured by roll-to-roll extrusion of plastic films over a master drum. Master drums are fabricated by spiral ruling on an ultraprecision drum lathe as shown in Figure 2 [2]. During ruling, copper plated drums are rotated at a slow rpm. A sharp diamond tool is placed on the linear axes and the motion is synchronized while the tool rules the vee-shaped pattern. Figure 3 is a diagram of the typical tool path during spiral ruling. The drum rotates while the tool traverses a loop to achieve the angled vee-groove. Patterns with various angles and grooves widths

are possible by adjusting the rpm of the spindle supporting the drum and the feed rate of the linear axes.



Figure 2. Moore Nanotechnology Systems HDL2000 drum lathe.

As with other optical components, surface finish and form of the vee-grooves is critical to optical performance. The grooves must be of equal width and depth throughout the entire drum. The most challenging feature is the width of the last groove because it must align perfectly to the first groove to maintain accurate groove width. Achieving accuracy requires a stable, stiff machine tool. The force on the sharp diamond tool is high and the cycle times can be very long. For instance, a 1 meter long drum, 300 mm in diameter would require up to 16,000 grooves. If each groove took 13 seconds, the total machining time is 60 hours.

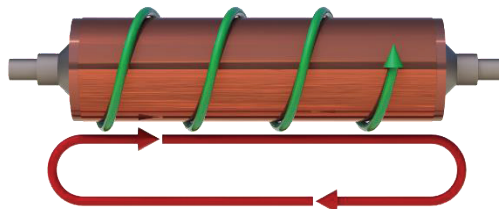


Figure 3. Diagram of tool path for spiral ruling of vee-groove patterns

Industry requirements of high production and sub-micrometer groove-to-groove spacing accuracy conflict to create a unique precision engineering challenge. This paper will cover the precision engineering solutions used to achieve fine pitch patterns on copper plated drums. Topics include; machine design, thermal

management of the machine and supply fluids and tool path optimization through modeling techniques.

MACHINE DESIGN

Optical surface finish and form are difficult to achieve by mechanical cutting. The machine tool must be rigid and isolated from external vibrations. Also, motion of the linear axes must be precise and accurate. Stiffness and damping is paramount due to the heavy chip load during ruling. The vee-groove pattern must be cut in one pass to minimize cycle time. Depths of cut up to 100 micrometer are required with minimal deflection to the tool with respect to the drum.

The HDL 2000 drum lathe employs oil hydrostatic bearings and direct-drive linear motors, see Figure 4. Glass scale encoders are used for 34 picometer feedback resolution. The combination of these technologies achieves high stiffness and precision, permitting linear feed rates up to 20 meter per minute and accelerations up to 0.05 g's. The headstock and tailstock spindles are hydrostatic, as well. The headstock spindle is equipped with a high torque motor and glass scale encoder. The spindle can be rotated to 300 rpm and position precisely at a resolution of 0.02 arc-seconds.

The HDL2000 has been design with a natural granite inertia block suspended on 8 isolators, as shown in Figure 5. Vibration isolators are used to achieve isolation from the ground. A welded steel frame raises the height of the isolators close to the cg of the machine, which minimizes motion of the granite during payload displacement. The tool path shown in Figure 3 results in a high acceleration of the linear axes (payload) therefore the location of the isolators is important to minimizing granite motion.

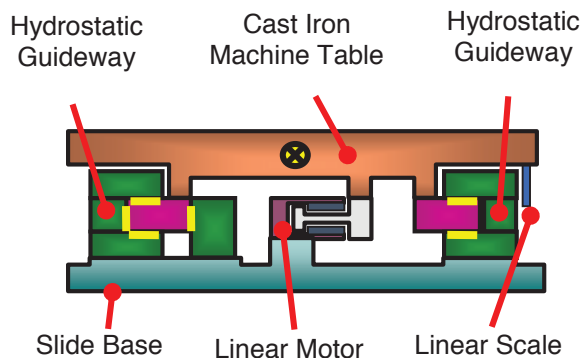


Figure 4. Drawing of the X-axis hydrostatic slide design utilized on the HDL2000.

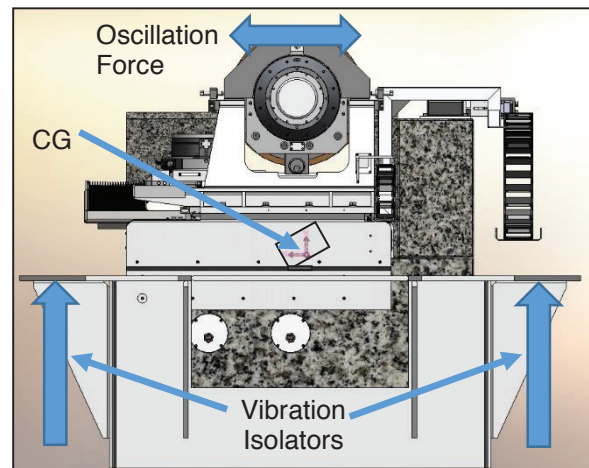


Figure 5. Diagram of the HDL 2000 drum lathe vibration isolation system.

THERMAL MANAGEMENT

As discussed previously, the cycle times in ruling can be many hours or days. Thermal stability of the entire machine, including the drum, is critical to holding sub-micrometer tolerances. A 1 degree Celsius change in the temperature of the drum equates to a 3 micrometer change in radius for a 250 mm radius drum. To achieve sub-micrometer precision it is imperative to maintain ± 0.1 degree Celsius temperature stability of all components. The HDL2000 is equipped with a temperature controlled enclosure that houses the entire machine, see Figure 6. The temperature of the enclosure is controlled with 2 HVAC units that feed air socks inside the enclosure. The HVAC units are controlled with a PID loop with temperature feedback from probes strategically suspended in the enclosure.



Figure 6. Picture of the temperature controlled enclosure for the HDL 2000 drum lathe.

All oil fed to the hydrostatic bearings is carefully temperature controlled. The oil is heated through a combination of flow and viscous shear of the film in the spindles and linear axes. The heat is removed from the oil using a refrigerated chiller and heat exchanger before re-circulating back to the bearings. Figure 7 is a plot of the temperature of several critical components during a ruling cycle. The first period of the cycle does not engage the drum and is intended to warm-up the bearings. The linear rails and motors build heat that is not detected in the returning oil. The duration of the required warm-up cycle will depend on the specific cycle, including the length of the drum and the axes feed rates.

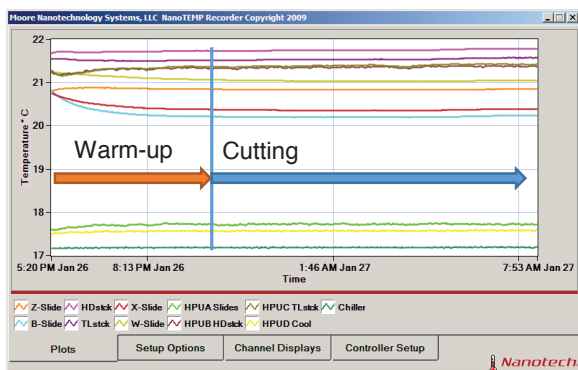


Figure 7. Temperature plot of critical components during ruling of a drum. Warm-up cycle is highlighted.

TOOL PATH DESIGN

Designing a tool path for drum ruling is challenging. The tool path must minimize cycle time and maximize precision. The techniques for achieving these goals are described in the next three sections.

Spindle Rotation at Constant Speed

The drum has significant mass and inertia and is therefore difficult to accelerate and decelerate. Following error of the spindle axis will increase with higher acceleration, which will increase form error of the grooves. Also, the spindle motor temperature will increase and affect spindle position during a long duration ruling cycle. The ideal tool path for the spindle axis is rotation at a constant speed, eliminating acceleration. The tool path is programed with spindle axis and linear axis positions on each line of the program. To achieve constant spindle axis rotation the feed rate is specified in inverse time and the c-axis increment is constant throughout the entire program. As the rotary axis rotates the linear axis

must be perfectly synchronized to maintain groove spacing.

Maximize Linear Feed Rate

Minimizing cycle time is achieved by maximizing feed rate. The quality of the machined surface does not vary with feed rate when ruling copper. Maximum feed rate should be used when possible. An algorithm is developed for tool path optimization to achieve synchronization of all axes while maximizing linear, vibration free feed rate. First the loop is divided into four sections; engage, cutting, retract, and return. Next, the cutting segment and the return segment are maximized for feed rate. Finally, the remaining time is allocated to the engage and retract profiles. See Figure 8 for a plot of the four segments.

Limit Acceleration

Acceleration rates of the linear axes must not exceed the limit of the machine. Exceeding the limit will cause the machine controller to slow all axes, including the rotary axis. Slowing the rotary axis causes acceleration or deceleration, therefore negating the advantages of rotation at constant speed.

The maximum acceleration of both the Z and X axes is 0.05 g's. During the engage and retract, the Z axis (cutting direction) must decelerate to a stop and accelerate to speed, which may be different for the cutting segment and the return segment. The X axis must accelerate from a stop and then decelerate to a stop. The profile of X versus Z displacement is calculated using a second order polynomial with the above constraints and equations placed in a system of equations.

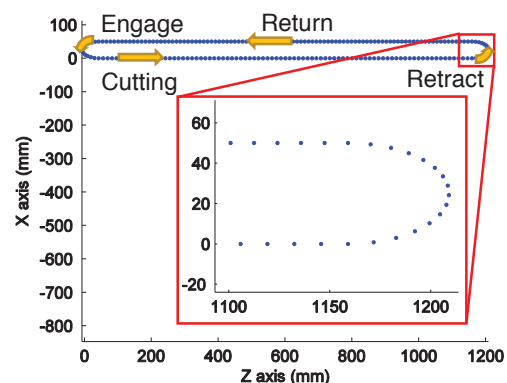


Figure 8. Plot of linear axis tool path during spiral ruling.

See Figure 8 for an example of point spacing used to maintain constant spindle rotation speed while minimizing acceleration. Each point represents a line of code in the control program. Each line has a C, X and Z axis value and the C axis (spindle axis) is incremented in equal increments. The respective acceleration plot is shown in Figure 9. As seen from the plots, the loop is traversed in 13 seconds while acceleration for the X and Z axes does not exceed 0.05 g's.

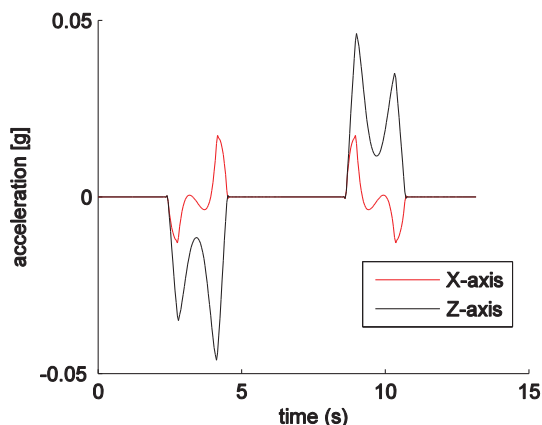


Figure 9. Plot of acceleration for one loop in the ruling cycle.

Software has been created to take inputs and automatically generate machine code with the above constraints. The user interface shown in Figure 10 can be used to generate loops stitched together in a program millions of lines long with generation only taking seconds. These programs permit entire drums to be machined with a seamless program that is easy for the machine controller to interpret.

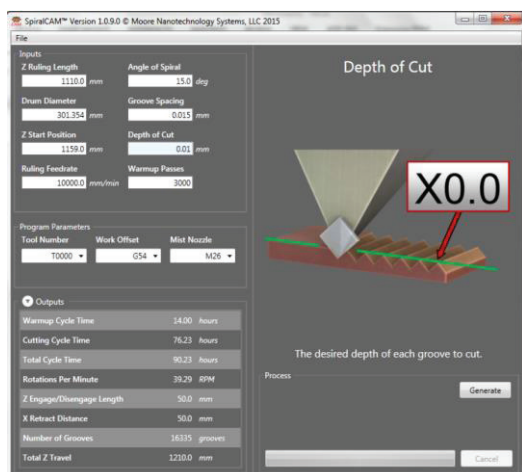


Figure 10. Picture of software interface to generate machine code for spiral ruling.

RESULTS

An example of a 15 micrometer vee-groove pattern on a copper drum is shown in Figure 11. The pattern was ruled with a 90° included angle tool tilted 15° from vertical. The pattern contained 16,335 grooves and the machining time was 60 hours. If the grooves were laid end-to-end they would span 70 km. The vee-grooves are made in 1 single cut at full depth.

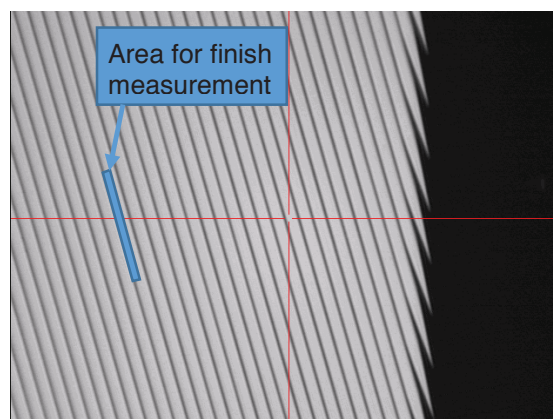


Figure 11. Microscope image of vee-grooves machined in copper by diamond turning. 15 μ m pitch, 15° from vertical.

The pattern in Figure 11 shows excellent groove spacing accuracy and includes the first and last groove. The entrance of the grooves on the drum is shown on the right side of the images. The tip of the tool creates a sharp point. The sharp point is maintained throughout the entire cut and illustrates tool wear is not an issue.

Surface roughness measurements of the vee-groove sidewall were taken by replicating the structure with a compound and measuring the replication on a white light interferometer. The resulting surface map shown in Figure 12. The surface is free from chatter or defects that would be optically perceptible. Also, the diamond tool edge quality is very good and does not cause striations along the length of the cut.

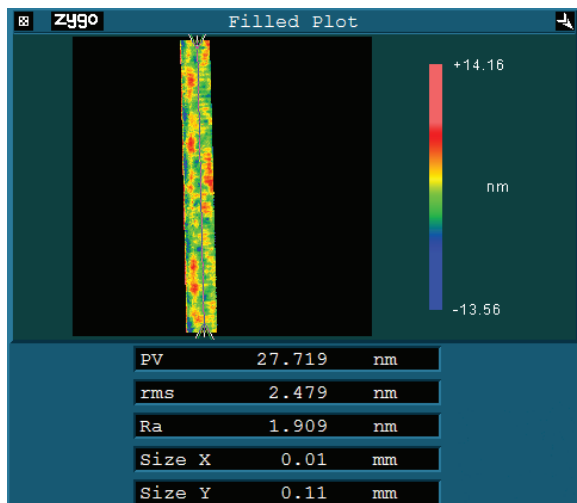


Figure 12. Surface map of vee-groove fabricated by spiral ruling with surface roughness parameters.

CONCLUSION

The HDL 2000 machine tool combined with an optimized tool path algorithm achieved fine-pitch vee-groove patterns. The form and finish of all grooves were of optical quality and met the specifications for quantum dot BEF films. Future work will expand the types of patterns to pyramidal and spherical patterns.

REFERENCES

- [1] J. Chen, V. Hardev, and J. Yurek, "Quantum Dot Displays: Giving LCDs a Competitive Edge through Color," *Nanotechnol. Law Bus.*, vol. 11, p. 4, 2014.
- [2] Dan Luttrell, "Innovations in Ultra-Precision Machine Tools: Design and Applications," in *tournament academic lecture Conference Paper Collection*, 2010, vol. 2, pp. 10–13.

LARGE-AREA, HIGH-PERFORMANCE FUNCTIONAL METROLOGY SYSTEM FOR NANOPHOTONIC DEVICE MANUFACTURING

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1. ABSTRACT

As cost-effective scalable nanofabrication techniques become available for nano-scale photonic devices, there is a need for large area metrology techniques that are preferably inline. These techniques are needed not only to establish/qualify manufacturing processes, but also to monitor process reliability for yield management. Here, we discuss the development of such a large area optical metrology system for nanophotonic devices – the nanophotonic functional metrology (NFM) system. The goal of this research is to design the NFM system with the following key features that are relevant in a manufacturing environment: (i) **Versatility** to allow the measurement of a variety of nanophotonic devices including wire-grid polarizers (WGPs) for LCDs [1-3], metal mesh transparent conductors for flexible displays and touch screens [4-6], plasmonic color filters [7-8], light-trapping structures for photovoltaics [9-11], and various kinds of novel metamaterials and photonic crystal structures [12-15]; (ii) **Large measurement range** to allow measurement over at least 6 decades of light intensity as theoretical contrast ratios for WGPs can extend upwards of 10^6 ; (iii) **High resolution** to detect sub-1% variations in functionality to enable high-resolution comparison between devices and point-to-point characterization across large area devices; (iv) **Reconfigurability** to enable customized automation of data collection at high speeds without compromising the above features of versatility, large measurement range, and high resolution. A first generation of this NFM system has been fabricated and its current functionality is illustrated for a high-contrast ratio WGP.

2. INTRODUCTION

Recent advances in scalable nanofabrication [1,16,17] is creating an interest in manufacturing of a variety of conventional and novel nano-scale photonic devices. Relevant devices include (i) High contrast ratio wire-grid polarizers (WGPs) for LCDs [1-3] that have the potential to improve

light management in displays leading to about 20% improved power consumption in LCDs; (ii) Transparent metal mesh films that form a flexible alternative to transparent conducting oxide layers with applications in flexible touch screens and displays [4-6]; (iii) Nanoscale patterned plasmonic color filters that can simplify fabrication of color filters for displays and cameras into a single patterned layer [7-8]; (iv) High efficiency light-trapping surfaces [9-11] for photovoltaics; (v) Metamaterials with unconventional optical properties such as negative index of refraction that enable ultra-high efficiency polarizers and novel diffractive light focusing optics [12-13]; (vi) Photonic crystal structures for enhanced light extraction from high index optoelectronic materials such as GaN [14-15]. Optical metrology is needed to qualify these kinds of devices as they come off the manufacturing line and to enable the investigation of advanced yield management strategies.

The devices themselves drive the required performance metrics of the metrology. For instance, theoretical contrast ratios for WGPs can extend upwards of 10^6 [1-3] requiring the instrument to be able to measure over 6 decades of light intensity. Also, haze (ratio of large-angle scattered transmittance to total transmittance) in metal meshes needs to be on the order of a few percent to compete with ITO [18], requiring the system to have enough sensitivity to detect sub-1% differences in device performance.

The NFM system not only needs to meet these performance specs but also needs to maintain a high level of versatility. For instance, WGPs typically only require measurements of total transmission, but as mentioned haze is an important metric for metal mesh which requires the measurement of scattered light. Furthermore, devices like light trapping structures are measured for reflectivity and both diffuse and specular reflectivity can be of interest. The NFM system must capture all these measurements.

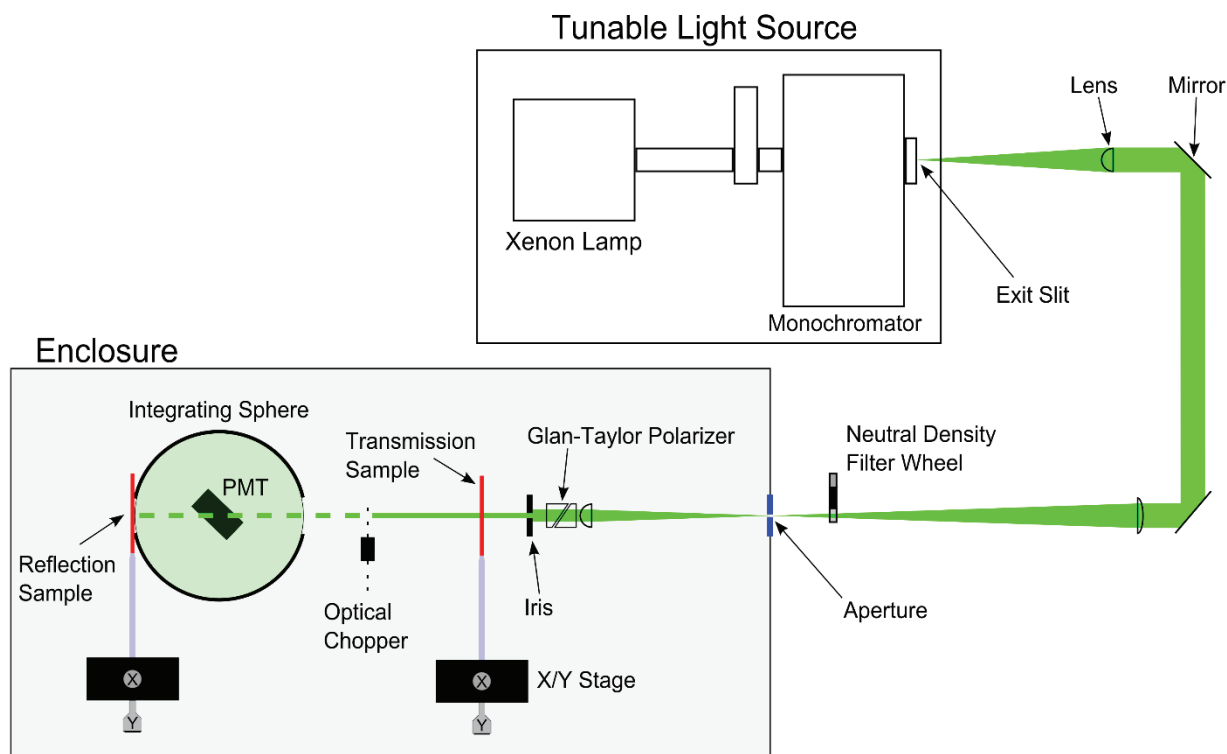


Figure 1. Schematic of the NFM System.

3. MOTIVATION OF THIS WORK

Commercial spectrophotometers achieve amazing specifications, routinely boasting dynamic ranges of beyond 8 decades and resolutions on the order of some tenths of a percent [19-22], but these systems are not ideal for high-throughput production of large-area nanophotonic devices. Most of these systems cannot be adapted for handling of large area substrates like wafers and roll-to-roll sheets. Furthermore, commercial systems are not highly configurable, both in terms of hardware and software, which makes them unsuitable for use in developing advanced metrology solutions for nanophotonic device manufacturing. This need has inspired the development of the custom optical metrology system called the Nanophotonic Functional Metrology (NFM) system.

4. NFM SYSTEM ARCHITECTURE AND DESIGN

4.1 NFM Architecture Overview

A schematic of the instrument is shown in Figure 1. The Xenon arc lamp emits from 250-2400 nm and is passed through a diffraction grating monochromator which can select a wavelength band with resolution from 0.7 to 42.1 nm

depending on the slit width. A lens is used to collimate the beam emerging from the monochromator and is turned with two mirrors. A second lens is then used to project the beam through neutral density filters (to control beam intensity) and onto a 1 mm circular aperture so that the beam becomes circular. Beyond this aperture the rest of the system is enclosed in a darkbox, the only opening in which is the aperture. The beam emerging from the aperture is collimated by a shorter focal length lens at a reduced diameter. The beam is then polarized with a glan-taylor polarizer, sent through an adjustable iris, and chopped with an optical chopper at around 1033 Hz. The beam then passes into the 6" integrating sphere. A photomultiplier tube (PMT) is attached to the bottom port of the integrating sphere to sense the light intensity. The output current from the PMT is sent across a 50 Ω load and a lock-in amplifier is used to read the resulting voltage. Samples are currently being positioned manually by micrometer driven X/Y stages which have 1" of travel on each axis and minimum step increments of 1/1000". Depending on the functionality of the nanophotonic device, samples can either be placed before the integrating sphere for transmittance measurements or at the reflection

port on the backside of the sphere for reflectance measurements.

With an 8 nm spectral bandwidth and a 1 cm beam diameter the NFM system can measure over 7 decades of light intensity and with a 1 mm beam diameter over 5 decades. It must be noted that the dynamic range of the system is much smaller than this (about 30000), but by known attenuation of the reference measurement much higher measurement ranges are achieved.

4.2 NFM System Design Theory

The NFM system is designed not only to be high performance, but also highly versatile. This is necessary because the NFM system needs to be able to measure a myriad of different devices each with their own specific functional performance metrics which span almost every conceivable photometric measurement.

The integrating sphere allows the NFM system to measure a near-exhaustive list of metrics including large-angle scattered transmittance, total transmittance, large-angle scattered reflectance, and total reflectance. Unscattered transmittance/reflectance can be easily back-calculated from these measurements. The integrating sphere's ability to measure large-angle scattered light in both reflection and transmission modes make it a necessary part of the NFM system. This enables both haze characterization in metal mesh devices and measurements of diffuse reflectance which may be important to light trapping structures. Unfortunately the integrating sphere reduces light throughput so it has a negative impact on the total measurement range that the system can achieve. However, if this became a limiting factor in the measurement of a device, the NFM system could readily be reconfigured to have the beam be directly incident on the PMT. This may actually become necessary if the contrast ratios of WGP's become large enough in the future.

The NFM system is not meant to be a one-size-fits-all metrology solution, but rather a system that can be carefully optimized for the needs of specific devices. Because the needs of WGP's and metal meshes are so different for instance, the ideal NFM system for each device would take on a significantly different form. Being a lab system, this version of the NFM system is meant to be highly reconfigurable, but commercially deployed NFM systems would most likely be configured with specific devices in mind. One of

the goals of this work is to determine the ideal NFM system configurations for specific classes of devices.

Commercial systems meet very impressive performance specs, but they are not readily adaptable to nanophotonic devices in a manufacturing environment. First of all, nanophotonic devices can be relatively large. Display devices like WGP's and metal meshes in particular can be manufactured on substrates ranging anywhere in size from smartphones to television displays. Even a smartphone display may be too large to fit inside most commercial spectrophotometers and certainly a device the size of a television would not. Furthermore, although basic automated sample handling is possible in commercial systems it is not readily extendable to high-throughput manufacturing scenarios which will demand customizable hardware and software.

4.3 Synchronous Detection

The NFM system utilizes synchronous detection to tolerate non-minimized stray light levels and extend the dynamic range of the system. In synchronous light detection the light beam is modulated by quickly blocking/unblocking the beam at a high rate using an optical chopper. A lock-in amplifier is then used to isolate the signal being modulated at the chopping frequency.

Synchronous detection is extremely well-suited to the current NFM system. Being that the instrument is expected to be reconfigured often, stray light control is not expected to be optimized. Synchronous detection allows for isolation of the beam from room lighting and scattered light from the optics inside the instrument. Interestingly enough, one of the most significant sources of stray light will come from the WGP's that are being tested which is the reflection of s-polarized light from the WGP. This can be directed into a light trap, but ideal light traps are not feasible, and backscatter from the light trap could still effect the accuracy of the measurement. Synchronous detection allows the NFM system to tolerate such noise as long as it is minimized with other optical design considerations.

4.4 Point-to-Point Scanning and Yield Management Strategies

The NFM system not only allows for the qualification of nanophotonic devices, but also lends itself the study of the relationship between spectrophotometric signatures and nanoscale

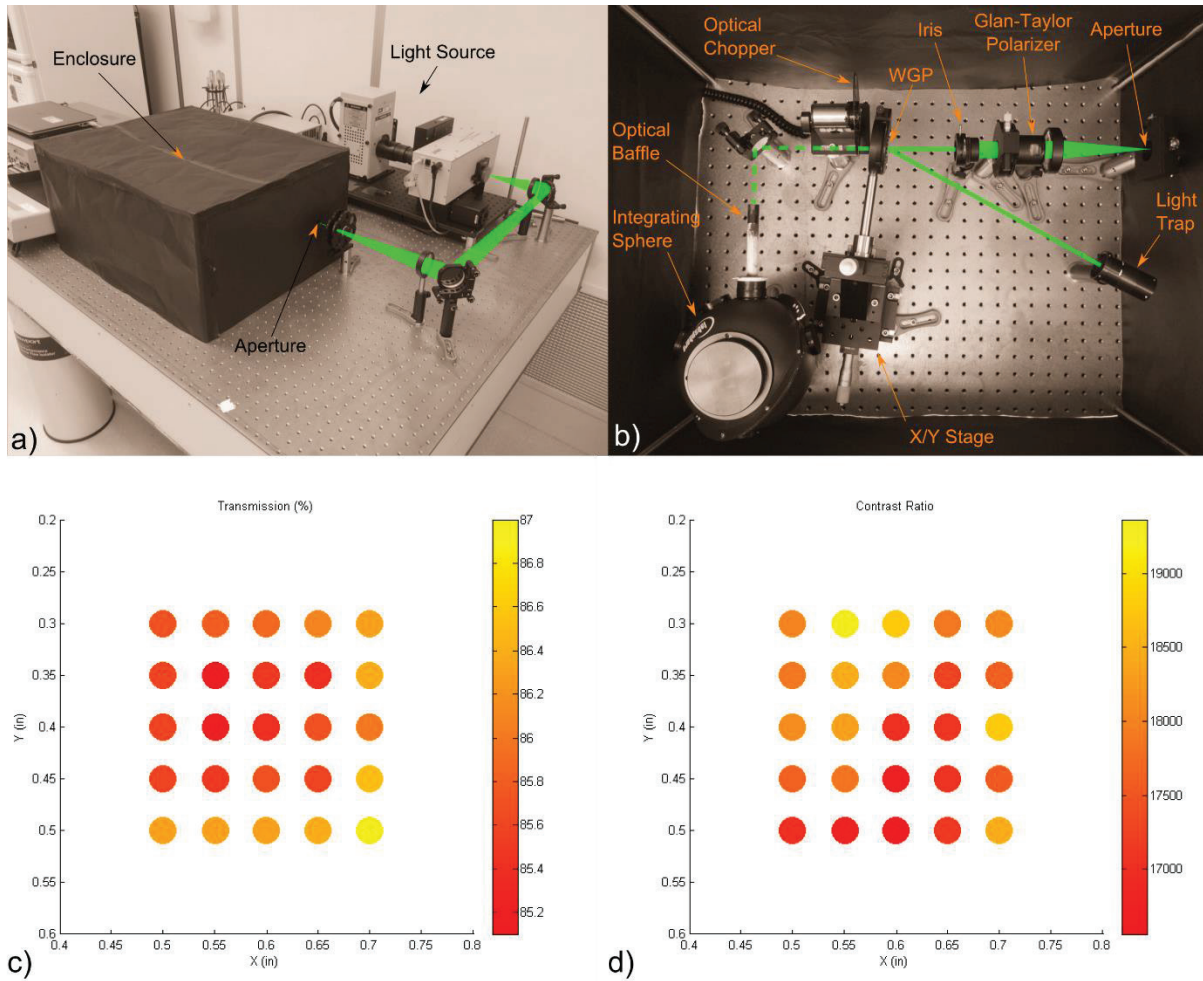


FIGURE 2. (a) Picture of the NFM system. (b) Picture taken inside the enclosure showing measurement configuration of a WGP. (c) Transmission (%) of p-polarized light through a WGP at 25 different points in a 0.2" x 0.2" area. (d) The corresponding contrast ratios of these 25 points. The beam size used for this experiment was ~ 1 mm. Measurements were done near the middle of a 1" x 1" sample, hence the offset of the plot axes.

pattern faults/geometric variability. Using small beam sizes (less than 1/100 of total device area) localized areas can be probed individually and compared in search for functional variations within and between devices. Certain types of pattern faults like feature collapse and missing pattern and geometric variabilities like differences in sidewall angle and critical dimensions may exhibit certain measurement signatures which when observed on the manufacturing line can be attributed to specific manufacturing steps involved in the fabrication of a device. This knowledge will be extremely useful for developing yield management strategies in factories producing nanophotonic devices on large scale. The NFM system enables the investigation of this topic.

5. RESULTS

Here we present the results from a high-end WGP involving fabrication of Aluminum line gratings on a glass substrate using Jet and Flash Imprint Lithography [3]. We measured the performance of the resulting high-contrast ratio WGP with the NFM system. Figure 2 shows 25 local measurements in a 0.2" x 0.2" area of the device. Transmission of p-polarized light (Figure 2c) and contrast ratio (Figure 2d) are shown. You can clearly see subtle variations in functionality between different locations on the WGP which can be attributable to either pattern faults, geometric variability, substrate inhomogeneity, and/or particle contamination. We plan to try to determine the exact cause of the variations in the future which could then imply a yield

management strategy for other WGP's being produced on the same manufacturing line.

Measuring the WGP successfully required careful arrangement of the NFM setup. As an example, originally the chopper was being placed before the sample, and measurements were clearly being skewed significantly by stray light as the contrast ratio was coming out more than an order of magnitude lower than expected. To fix this issue the chopper was placed after the sample and the reflection of s-polarized light from the WGP was directed into a light trap (Figure 2b). In this configuration only the transmitted light was chopped and thus the reflected light from the WGP did not affect the signal reading. Furthermore an optical baffle was extended from the integrating sphere input port to further prevent stray light from reaching the PMT (Figure 2b).

This measurement was limited to such a small area because the current system is being manually operated. The system can be upgraded to automatic control – which is our plan – and at that point the data collection abilities of the system will increase dramatically. For instance, with a spot size of 1 mm, a 1 in2 area can contain 625 individual local measurements. Larger spot sizes can be used for larger samples.

6. CONCLUSIONS

A highly configurable, high-performance optical metrology system – known here as the nanophotonic functional metrology (NFM) system – has been created for nanophotonic device manufacturability studies. The system can measure over 6 decades of light intensity and has the ability to detect sub-1% differences. The system's high level of configurability is an important figure of merit because it allows for the development of very specific metrology solutions which will depend heavily on the devices being measured. The NFM system allows for the qualification of nanophotonic devices like WGP's, metal meshes, plasmonic color filters, and light-trapping structures, and also for the investigation of the metrology signatures of pattern faults and geometric variability that arise from manufacturing parasitics that can affect process yields. Understanding of these metrology signatures can enable advanced yield management strategies in factories producing nanophotonic devices on large scales.

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REFERENCES

- [1] Ahn S.W, Lee K.D, Kim J.S, Kim S.H, Park J.D, Lee S.H, Yoon P.W. Fabrication of a 50 nm half-pitch wire grid polarizer using nanoimprint lithography. *Nanotechnology*. 2005;16:1874–1877.
- [2] Watts M.P.C, Little M, Egan E, Hochbaum A, Jones C, Stephansen S. A process for, and optical performance of, a low cost Wire Grid Polarizer.
- [3] Ahn S.H, Miller M, Yang S, Ganapathisubramanian M, Menezes M, Singh V, et al. High Volume Nanoscale Roll-based Imprinting Using Jet and Flash Imprint Lithography. *SPIE*. 2014;9049.
- [4] Wu H, Kong D, Ruan Z, Hsu P.C, Wang S, Yu Z. A transparent electrode based on a metal nanotrough network. *Nature Nanotechnology*. 2013;8:421-425.
- [5] Guo C.F, Ren Z. Flexible transparent conductors based on metal nanowire networks. *Materials Today*. 2015;18:143-154.
- [6] Hu L, Wu H, Cui Y. Metal nanogrids, nanowires, and nanofibers for transparent electrodes. *MRS Bulletin*. 2011; 36:760-765.
- [7] Xu T, Wu Y.K, Luo X, Guo L.J. Plasmonic Nanoresonators for High-Resolution Colour Filtering and Spectral Imaging. *Nature Communications*. 2010;1.
- [8] Yokogawa S, Burgos S.P, Atwater H.A. Plasmonic Color Filters for CMOS Image Sensor Applications. *Nano Lett*. 2012; 12:4349–4354.
- [9] Brongersma M.L, Cui Y, Fan S. Light management for photovoltaics using high-

- index nanostructures. *Nature Materials*. 2014;13:451-460.
- [10] Priolo F, Gregorkiewicz T, Galli M, Krauss T.F. Silicon Nanostructures for Photonics and Photovoltaics. *Nature Nanotechnology*. 2014;9:19-32.
- [11] Hilali M, Sreenivasan S.V. Nanostructured Silicon-Based Photovoltaic Cells. *High-Efficiency Solar Cells Physics, Materials, and Devices*; Editors: Xiaodong Wang, Zhiming M. Wang; Springer Series in Materials Science. 2014;190.
- [12] Monticone F, Alù A. Metamaterial-Enhanced Nanophotonics. *OPN - Year in Optics*. 2013;24(12):35.
- [13] Shen B, Wang P, Polson R, Menon R. An ultra-high efficiency metamaterial polarizer. *Optica*. 2014;1(5):356-360.
- [14] Wierer Jr. J.J, David A, Megens M.M. III-nitride photonic-crystal light-emitting diodes with high extraction efficiency. *Nature Photonics*. 2009;3:163-169.
- [15] Hershey R, Miller M, Jones C, Subramanian M.G, Xiaoming L, Doyle G, Lentz D, LaBrake D. Nanophotonics Foresight Report. *Proc. SPIE Microlith*. 2006;6337.
- [16] Sreenivasan S.V. Nano-Scale Manufacturing Enabled by Imprint Lithography. *MRS Bulletin, Special Issue on Nanostructured Materials in Information Storage*. 2008;33:854-863.
- [17] Choi B.J, Schumaker P.D, Xu F, Sreenivasan S.V. UV Nanoimprint Lithography. *Handbook Of Nanofabrication*. 2009 October:149-181.
- [18] Colin Preston, Yunlu Xu, Xiaogang Han, Jeremy N. Munday, and Liangbing Hu, Optical haze of transparent and conductive silver nanowire films, *Nano Research* 2013, 6(7): 461–468
- [19] Agilent Technologies. Agilent Cary 4000/5000/6000i Series UV-VIS-NIR. https://www.agilent.com/cs/library/brochures/5990-7786EN_Cary-4000-5000-6000i-UV-Vis-NIR_Brochure.pdf.
- [20] JASCO. V-700 series UV-Visible/NIR Spectrophotometers. <http://www.jascoinc.com/spectroscopydownload-catalog?category=uv-visible-nir>.
- [21] SHIMADZU. SolidSpec-3700 SolidSpec-3700DUV; Shimadzu UV-VIS-NIR Spectrophotometer. http://www.ssi.shimadzu.com/products/literature/spectroscopy/shimadzu_solidspec_bf.pdf.
- [22] PerkinElmer. LAMBDA UV/Vis and UV/Vis/NIR Spectrophotometers. http://www.perkinelmer.com/CMSResources/Images/4474450BRO_LAMBDA8509501050.pdf.

FLEXIBLE GLASS SUBSTRATES FOR ROLL-TO-ROLL MANUFACTURING

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ABSTRACT

As optical and electronic devices evolve to become lighter, thinner, and more flexible, the substrate choice continues to be critical. The substrate directly affects improvements in the designs, materials, fabrication processes, and performance of advanced electronics. With their benefits such as surface quality, optical transmission, hermeticity, and thermal and dimensional stability; glass substrates enable high-quality and long-life devices. As substrate thicknesses are reduced below 200 μm , ultra-slim flexible glass continues to provide these benefits to high-performance flexible devices. In addition, the reduction in glass thickness also allows for new device designs and high-throughput, continuous manufacturing enabled by roll-to-roll (R2R) processes. This paper provides an overview of flexible glass substrates and how they enable high quality roll-to-roll manufacturing.

Flexible Glass

With the continuing emergence of new printed electronic applications, device designs, and processes; there is an increased focus on optimized material selection. As part of this, the device substrate plays a critical role in device manufacturing and end-use performance. During device manufacturing, the substrate influences the overall process parameter ranges as well as the resulting feature resolution and layer-to-layer registration. During end-use application, the substrate affects device performance and lifetime. As printed electronic devices advance and become more integrated, the substrate choice is critical.

Flexible glass substrates have several advantages that enable high-performance

optical and electronic devices.[1] With thicknesses in general $\leq 200\mu\text{m}$, flexible glass has lower stiffness and bend stress compared to rigid glass substrates. This enables use of flexible glass in R2R device fabrication and conformable device applications. Compared to alternative flexible substrate materials such as polymer film and metal foil, glass has advantages of optical quality, surface quality, thermal and dimensional stability, process compatibility, and hermeticity. Following are some highlighted examples:

- Optical transmission: 100 μm thick glass has a flat optical transmission $>350\text{nm}$ through the visible and NIR spectrum. In this range there is negligible haze or absorption.
- Surface roughness: Over a 300 μm x 300 μm window, the flexible glass has a $R_a < 0.5\text{nm}$
- Dimensional stability: No detectable dimensional distortion is measured in flexible glass substrates during thermal cycling typical of printed electronic processing or 85°C/85% rh damp heat exposure.
- Hermeticity: The WVTR of flexible glass is below the $10^{-7} \text{g/m}^2/\text{day}$ detection limit of the most sensitive system used to-date.

Flexible Glass R2R Processing

Similar to other flexible substrate materials such as polymer film, paper, and metal foil; the most efficient way of handling large areas of self-supported flexible glass is with roller system conveyance. To highlight the compatibility of flexible glass web with roller conveyance, R2R demonstrations were performed in key building-block processes. These processes can be used individually to form or pattern single layers on flexible glass web, or they can be combined to

fabricate complete functional devices in an integrated R2R manufacturing approach. Figure 1 shows a glass spool with a width >1m as well as flexible glass web conveyance that enable R2R device processing. Efficient use of flexible glass web has been demonstrated in R2R processes such as: thin film vacuum deposition, polymer film lamination, photolithography, laser ablation patterning, slot die coating, and printing methods.[2-5]



FIGURE 1. Flexible glass web and R2R conveyance.

To highlight the mechanical reliability of flexible glass, both bend testing of discrete substrates as well as repeated glass web conveyance trials were performed. Cyclic bend testing at radii $\leq 50\text{mm}$ was successfully performed up to 25,000 cycles in order to evaluate device layers deposited on the flexible glass.[6] Similarly, repeated glass web conveyance trials were performed over 30 times demonstrating that the glass did not accumulate strength-limiting defects during metal roller conveyance.[7] These R2R process trials as well as mechanical reliability evaluations indicate the compatibility of flexible glass web with R2R device manufacturing processes.

Flexible Glass Printed Electronics

Previous work has described the compatibility of flexible glass with solution-based processing in general and printed electronics specifically.[8] This has included methods for glass cleaning to create a high surface energy condition ($>75\text{mJ/m}^2$) for appropriate ink wetting. Previous solution-based processing demonstrations have included: slot die coating, screen printing of features down to $30\mu\text{m}$, gravure printing, gravure-offset printing, and ink jet printing. Flexible glass devices fabricated with solution-based processing have included organic photovoltaics and organic TFTs.

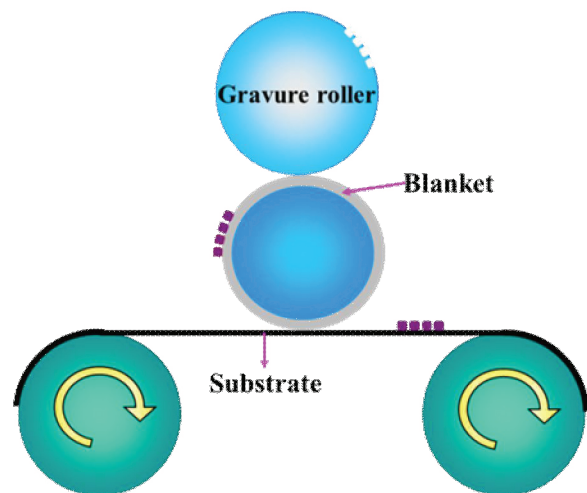


FIGURE 2. R2R gravure-offset printing.

This paper specifically highlights the recent advances in use of flexible glass in R2R and sheet-based gravure-offset printing. Figure 2 shows a gravure-offset process diagram where ink on the engraved gravure cylinder is transferred to the blanket roller and then to the glass web substrate.

In terms of general printing capability, Figure 3 shows examples of Ag-ink gravure-offset printed patterns on flexible glass with linewidths down to $10\mu\text{m}$. Figure 4 shows flexible glass web in a R2R gravure-offset printing process. In this case, Ag-ink metal mesh patterns with linewidths $<20\mu\text{m}$ were printed on 500mm-width glass web. The optical transmission was $>80\%$ with sheet resistance $<40\text{ohm/sq}$. This demonstrated process is capable of a 100mm/s R2R line speed limited by the drying oven length, and a linewidth variation of less than $\pm 20\%$. Further improvements in optical transmission and sheet resistance may be possible with Ag-ink

composition and pattern design. A comparison of gravure-offset printed Ag-ink features on flexible glass and polymer substrate will be provided along with initial device demonstrations.

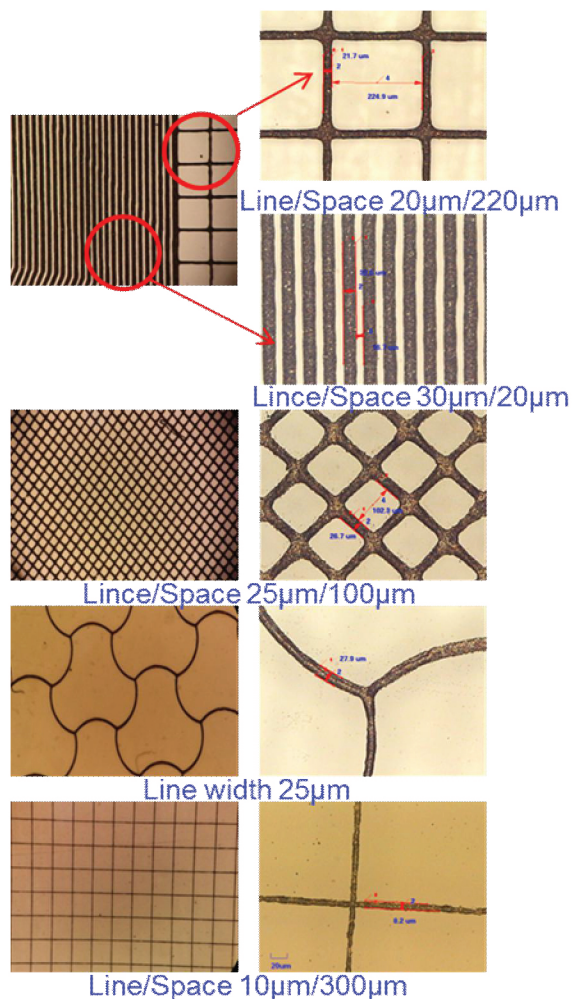


FIGURE 3. Gravure-offset printed Ag-ink features on flexible glass substrates.

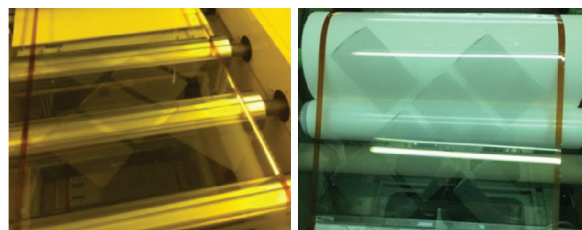


FIGURE 4. R2R gravure-offset printing of Ag-ink metal mesh on flexible glass web.

More specifically related to printed functional electronic device structures, FIGURE 5 shows a

gravure-offset printed 23-inch touch sensor pattern on 100 μm -thick flexible glass. This includes 70 μm interconnect lines as well as metal mesh patterns with an average width of 10.7 μm and deviation of 1.2 μm .

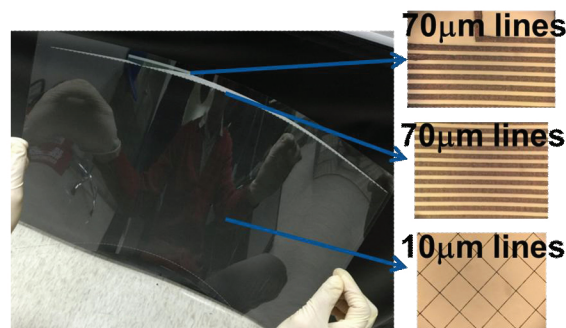


FIGURE 5. Gravure-offset printed 23-inch touch sensor on 100 μm -thick flexible glass.

As shown in Figure 4, 7-inch touch sensor patterns were printed by a R2R gravure-offset process on flexible glass web. This device was a single-layer, single-sided design with a linewidth of 13.6 $\mu\text{m} \pm 1.1\mu\text{m}$. The sheet resistance was 55 Ω/sq , and the optical transmission was 85% including losses due to surface reflection. Figure 6 shows that these devices can be cut from the overall flexible glass web and connectorized to the required driving board. Figure 7 shows the devices in both single-point and multiple-point operation.

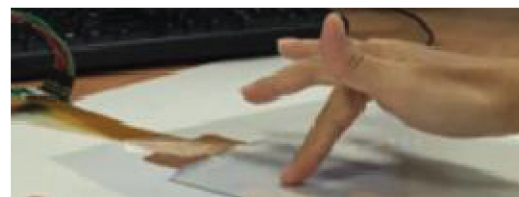


FIGURE 6. Singulated and connectorized flexible glass device.

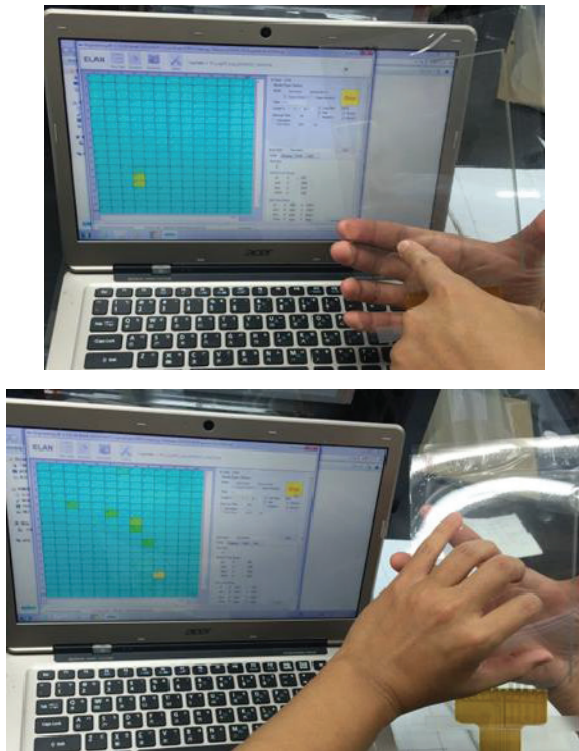


FIGURE 7. Single- and multi-point touch operation of flexible glass device fabricated by R2R gravure-offset printing.

Summary

Flexible glass offers several advantages for printed electronic applications such as optical quality, surface quality, thermal and dimensional stability, process compatibility, and hermeticity. The mechanical reliability of flexible glass web and compatibility in key R2R processes has also been demonstrated. Combining these with advances in gravure-offset printing methods on flexible glass enables advances in printed electronic device manufacturing as well as device performance. With the replacement of multi-step photolithography processes with a single continuous printing step, R2R gravure-offset printing on flexible glass web enables potentially game-changing flexible electronic manufacturing methods.

References

- [1] Garner S, Glaesemann S, Li X. Ultra-slim flexible glass for roll-to-roll electronic device fabrication. *Appl. Phys. A*. 2014; 116: 403-7.
- [2] Garner S, Merz G, Tosch J, Chang C, Lin J, Kuo C, et al. Ultra-Slim Flexible

Glass for Electronic Application. MRS Fall Meeting; 2012 Nov28; Boston.

- [3] Garner SM, Lin JC, Kuo KT, Tseng PL, Lin SM, Huang MH, et al. Flexible Glass Substrates for Printed Electronic Applications. LOPEC; 2015 Mar 2-5; Munich.
- [4] Garner S, Lin JC, Kuo KT, Li SM, Tseng PL, Huang MH, et al. Roll-to-Roll Vacuum Coating and Device Fabrication Using Flexible Glass Web. SVC TechCon; 2015 Apr 27-30; Santa Clara.
- [5] Poliks MD, Hewlett J, Malay R, Nandur A, Vaddi R, White B, et al. Active and Passive Integration on Flexible Glass Substrates: Subtractive Single Micron Metal Interposers and High Performance IGZO Thin Film Transistors. IEEE Electronic Components and Technology Conference; 2015 May 26-29; San Diego.
- [6] Burst JM, Rance WL, Meysing DM, Wolden CA, Metzger WK, Garner SM, et al. Performance of Transparent Conductors on Flexible Glass and Plastic Substrates. IEEE-PVSC; 2014 June 9-13; Denver.
- [7] Garner SM, Huang MH, Pollard S, Daly C, Lin JC, Kuo KT, et al. Ultra-Slim Flexible Glass for Optical and Electronic Applications. Electronics Packaging Symposium; 2014 Oct 8-9; Binghamton, NY.
- [8] Garner S, Merz G, Tosch J, Myers T, Boudreau R, Rebros M, et al. Glass Substrates for Printed Electronics. FlexTech Alliance Workshop: Printing Electronics-Ink and Substrate Interactions; 2012 Aug 2; Kalamazoo, MI.

MATERIAL DISPENSING OPTIONS FOR ROLL-TO-ROLL UV NANOIMPRINT LITHOGRAPHY

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ABSTRACT

Roll-to-roll UV Imprint Lithography (R2R-UVIL) is being developed at UT-Austin as an adaptation of the wafer-based UV nanoimprint process. The R2R-UVIL is targeted to achieve high throughput (>20ft/min) sub-50nm patterning of a variety of feature types on flexible substrates. Material dispensing is a critical building block for this process. In the domain of wafer-based processing, spin-coating and inkjetting are typically the only viable material dispensing techniques that exist, of which inkjetting has demonstrated advantages due to its ability to handle complex patterns, and substantially lower material wastage. On the other hand, in the world of R2R processing, driven by the mature plastics, paper and fabric processing industries, several material dispensing techniques have been investigated and developed for micro-scale films. Some popular methods include slot-die coating, inkjetting, gravure printing, screen printing, etc. However, most of these techniques fall under the gamut of “printing” rather than “coating”, and are utilized best for dispensing material over smaller 2D areas rather than the entire surface of the flexible substrate. Given their ability to coat sub-micron films over large areas with high throughput and low waste, slot-die coating and inkjetting are the two most viable options for material dispensing for R2R-UVIL. While inkjetting has a tighter material rheology window decreasing material versatility, slot-die coating does not yield sub-100nm thick films with ease and also suffers from a dependence of the final film thickness on the initial gap between the coating head and the substrate. This paper attempts to quantify the above comparison in the context of process development for R2R-UVIL, based on preliminary models and experimental results. Both inkjetting and slot-die coating can yield nanoscale thin films, but there are additional constraints on using either modality. The ultimate choice finally depends on trade-offs between film thickness and uniformity, material versatility,

amount of material consumed including wastage, and overall process throughput.

1. INTRODUCTION

The ability to pattern materials at the nanoscale enables a wide variety of applications. Material deposition is a critical unit process for nanopatterning. The choice of material deposition technique influences both, the performance and economics of the overall nanopatterning process. Hence, it is important to understand the nature of nanoscale deposition processes and their compatibility with nanopatterning.

1.1 Nanopatterning Techniques

Nanopatterning techniques typically come in three flavors: (i) top-down, such as photolithography and nanoimprint lithography, (ii) bottom-up, such as block copolymer self-assembly, and (iii) hybrid, such as directed self-assembly processes [1]. For this work, top down nanoimprint lithography process has been considered and elaborated below.

1.1.1 Jet-and-Flash Imprint Lithography

Jet and Flash Imprint Lithography (J-FIL, cf. Figure 1) is a variant of UV-based nanoimprint lithography [1]. It uses inkjets to dispense drops of a UV-curable monomer onto a substrate. A template with the etched nanoscale features is then brought down on these drops such that it forces the drops to merge and form a contiguous fluid film while filling the nanoscale features. Once the features are filled, the template-monomer-substrate sandwich is flood exposed with UV light to crosslink the monomer and solidify it. Then, the template is separated leaving the nanostructured polymer film behind. This standard wafer-based process has also been modified for handling flexible substrates in a R2R configuration in a prototype tool capable of imprinting on 80mm wide flexible substrates. This tool does not use a continuous process, as would commonly be imagined for R2R processing. Instead, it leverages the same basic tool design

and template as wafer-scale J-FIL and imprints in a step-and-repeat mode on flexible substrates that are handled R2R. This modified substrate module is shown in Figure 1.

1.2 Nanoscale Film Deposition Techniques

Deposition of nanoscale films can be done in both the solution and vapor phases. Vapor phase deposition processes such as CVD, ALD, PVD, etc. require high vacuum and are not relevant in the context of R2R-UVIL configuration. Hence, the focus here is on deposition techniques that are solution-based. There are several deposition techniques compatible with R2R processing, driven by the mature plastics and paper industries. Of all the techniques, slot-die coating and inkjetting were chosen because of their superior characteristics in terms of minimum deposited film thickness, throughput, low material waste and choice of material rheology.

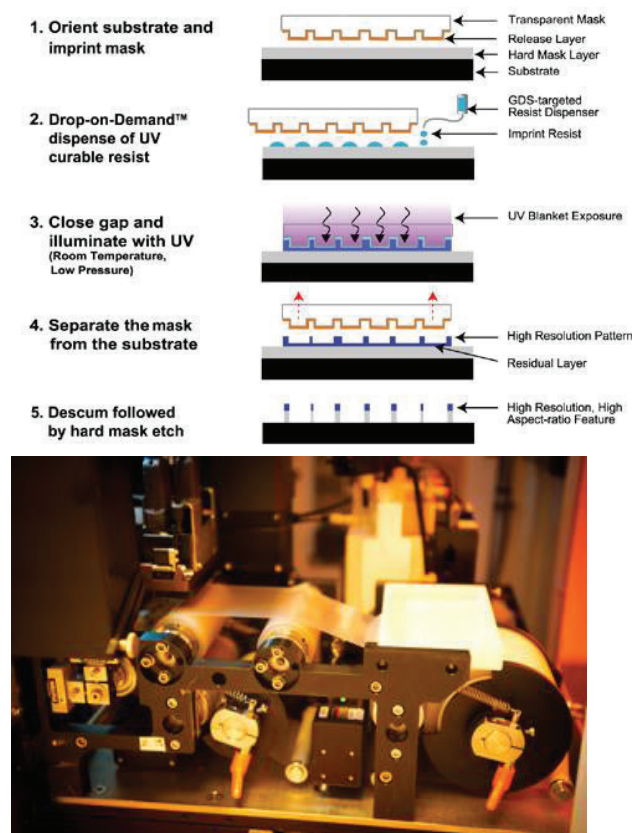


FIGURE 1. (Top) Illustration of the (J-FIL) process. (Courtesy: Molecular Imprints, Inc.), (Bottom) Picture of the flexible substrate module inside the R2R-JFIL prototype tool installed at the NASCENT Center.



FIGURE 2. A picture of the R2R slot-die coating tool installed at the NASCENT Center.

1.2.1 Slot-die Coating

Slot-die coating (see Figure 2) is a widely used process in several industries including coating for paper, plastics and photovoltaics. It belongs to a class of techniques called pre-metered coatings as the amount of liquid coated is metered by way of the volume of liquid pumped through the die which has a precisely machined slot that governs the formation of a coating bead. The process can be used deposit a wide variety of material rheologies in multiple configurations, including single-layer films, multi-layer stacks, and can be implemented on rigid substrates or in a roll-to-roll configuration. However, its relative difficulty to get to nanoscale thickness films creates challenges in its use as a material dispensing technique for nanoscale lithography.

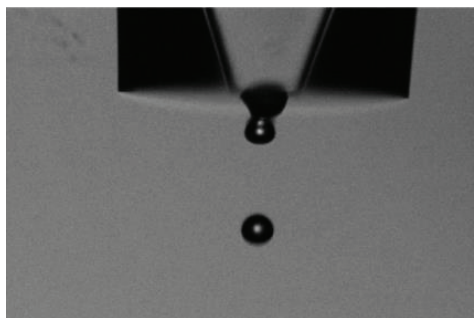


FIGURE 3. Screen grab of an inkjet nozzle dispensing drops of water.

1.2.2 Inkjetting

Inkjetting is a widely used material deposition technique. It has been reviewed extensively in the literature [2]. Inkjetting can be both continuous and drop-on-demand. While the former is used in commodity applications, the latter is more commonly used for nanoscale film deposition and also where excessive material consumption is not desirable. Drop-on-demand inkjets typically consist of one or more nozzles that are squeezed or expanded by the action of a piezoelectrically actuated pressure pulse. Precise tuning of the

pulse is needed to ensure reliable dispensing of drops of a given volume and velocity for an inkjet material with a specific rheology (Figure 3). Not all liquids are compatible with inkjetting, especially given that low volume drops (sub-10 pico liters) are needed to get to films of sub-100nm thickness.

1.3 Motivation

Choice of the appropriate material dispensing method is primarily driven by throughput, minimum feature resolution, minimum film thickness, film thickness uniformity, material compatibility and material wastage. The motivation is to compare slot-die coating and inkjetting on these metrics as much as possible to determine their feasibility for use in nanoscale imprint lithography.

2. INKJETTING PROCESS DEVELOPMENT

Inkjet technology has been studied extensively in the literature. However, when applied to the case of UV nanoimprint lithography, the inkjetted drops are forced to merge under the action of an overlying superstrate. The presence of this “infinitely rigid” superstrate acting on drops dispensed on a highly flexible substrate creates new flow behaviors that are important to characterize to understand a viable process window, especially for minimum film thickness and film thickness uniformity.

State-of-the-art inkjet volume resolution demonstrated in J-FIL is on the order of 1pl while the currently installed inkjetting system for R2R nanoimprint lithography has a volume resolution of 6pl, with a minimum volume of 6pl and maximum volume of 42pl. At the same time, it consists of >150 nozzles activated by the same head, with a lateral nozzle pitch of 84.5 μm . A preliminary modeling exercise has been undertaken to understand how the coercing action by the “infinitely rigid” superstrate affects drop spreading, merging and ultimately thickness uniformity of the resulting film on a flexible substrate. The model used to understand the inherent non-uniformity in the spreading process, is an extension of thin film lubrication model.

2.1 Model Development

Typically, a UV-curable imprint resist is highly wetting. Given this and the fact that the resist also has low contact angle on both substrate (<10 degrees) and template (~25 degrees), the radius of a drop (~100 μm) tends to be much larger than the height (<1 micron), validating the use of lubrication approximation to model thin film flow.

The basic physics is that of thin film fluid flow confined by one rigid (typically fused silica template in R2R patterning) and one elastic substrate (typically the flexible substrate that is being patterned). The spreading of individual drops can be construed as the driving of contact lines by the wetting characteristics of the fluid as well as the coercing of the superstrate, which is assumed to not have any nanopattern for the sake of model development. It must be noted that the merging process cannot be captured by this model, as it violates the basic lubrication assumption of negligible pressure gradients in the thickness direction. In practice if a rational drop dispense pattern consistent with local fluid requirements of the template is used, it has been found that any air trapped between the drops get absorbed into the liquid monomer and/or the flexible substrate within a few seconds due to the high capillary pressure in the trapped bubble.

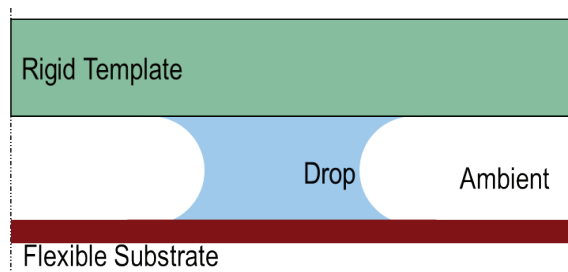


FIGURE 4. Schematic of the geometry used in the model and simulation.

Simulations have been conducted by assuming the drop volume as 6pl and drop pitch corresponding to multiples of the nozzle pitch on the inkjet which is 84.5 μm . The initial spread is calculated by assuming that the dispensed drop takes on the shape of a spherical cap with an approximate contact angle of 9 degrees on the substrate and 25 degrees on the substrate. The superstrate is assumed to be infinitely rigid and the substrate assumed to be held in tension at 1N/m. The influence of drop pitch is studied through numerical simulation of the system. The key output metrics include: (i) mean film thickness, (ii) % non-uniformity or the ratio of standard deviation with mean film thickness, and (iii) time taken for neighboring drops to merge. While the mean film thickness and standard deviation help in understanding the inherent non-uniformity post-spreading and pre-merging, the time taken for the drops to merge (pre-merging time) helps in understanding the minimum time that a drop takes to merge with its neighbor after

initial superstrate contact. A schematic of the initial geometry is given in Figure 4.

2.2 Model Results

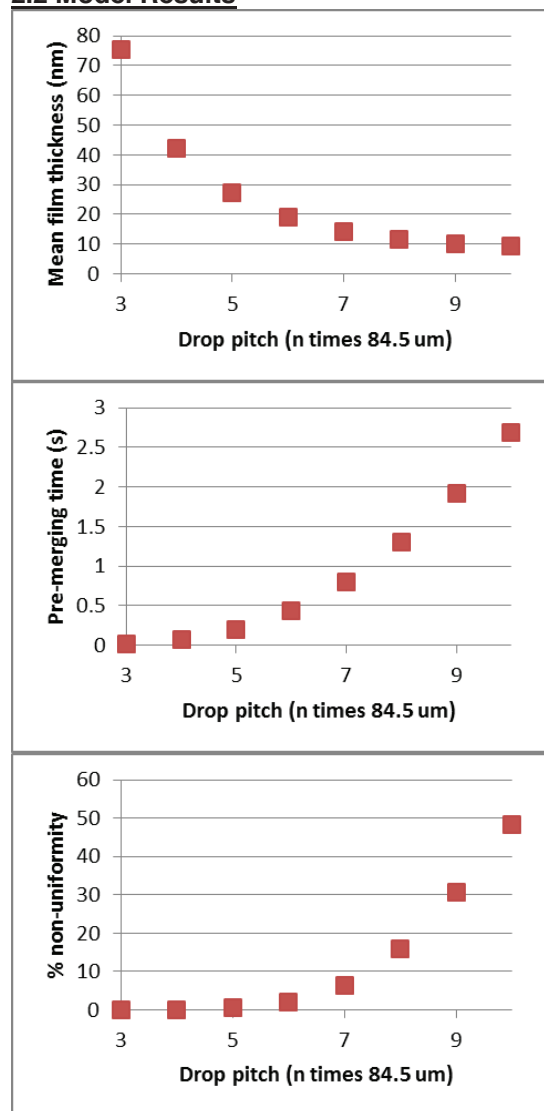


FIGURE 5. Model results showing the influence of drop pitch on (top) mean film thickness, (middle) time taken for neighboring drops to merge, and (bottom) % non-uniformity, measured as the ratio of standard deviation to mean film thickness.

From the plots in Figure 5, drops spaced farther apart result in thinner films and take longer to merge, which is expected. However, the % non-uniformity becomes a problem at sub-20nm films. This can be attributed to the fact that as the drops are spaced farther apart, they result in thinner films which drives the capillary pressure up. The increase in capillary pressure results in greater deformation of the flexible substrate and creates

a substantial deviation from uniformity while spreading. These results could not be exactly validated experimentally because of the lack of a thin film metrology tool capable of measuring nanoscale films on transparent flexible substrates. However, the same idea was replicated for rigid wafers on which the film thickness could be measured, and good qualitative agreement with model prediction was found.

3. SLOT-DIE COATING DEVELOPMENT

Like inkjetting, slot-die coating is also capable of high throughput, low wastage and high uniformity coatings. It generally is less restrictive in material rheology choice compared to inkjetting, but reaches nanoscale film thicknesses with difficulty. In its capabilities, it is complementary to inkjetting.

3.1 Machine Geometry

The slot-die coating tool installed at NASCENT Center is capable of coating on flexible substrates upto 350mm wide and that are about 150 μ m thick in a roll-to-roll configuration. In its current form, its coating width is limited to 210mm on both substrates. The tool consists of a syringe pump to push the material into a stainless steel die with accurately machined lips that maintain a gap of a few microns through which liquid is pushed out. The relative motion between the die and the substrate is maintained by moving the die and not the substrate. Because the die moves, the tool is not capable of performing a continuous coat, but can do so in individual sections that are chucked on a granite vacuum chuck. The coated sections are then passed through an oven for curing the coated liquid films and for vaporizing solvents used. For flexible substrates that come with a laminated protective layer, two handling modules have been custom designed – one that automatically takes off the protective film before coating and another that puts on a new protective film after the substrate has been processed through the oven. This allows some unique batch processing capability in which an entire roll, once coated, can be sent through another R2R tool, for example, nanoimprint, without damaging the coating while avoiding exposure to surface or particulate contamination.

The die also uses an indexing sensor to know the gap between the substrate and the lips before each coating. This is needed to maintain an accurate coating gap and to make sure that the lips do not run into the substrate. The tool is rated

to coat materials that have viscosity ranging from 1-100cps and surface tension from 10-50 mN/m.

3.2 Important Process Parameters

Slot-die coating relies on an optimal interplay of several process parameters to enable defect-free coating. Some important parameters on the machine side include relative speed between the substrate and die, gap between the die and the substrate and material pump rate. On the material side, important parameters include viscosity, surface tension, density and solids content. These parameters are not only important in establishing the coating window, they also influence the final film thickness that is obtained.

3.3 Coating Window

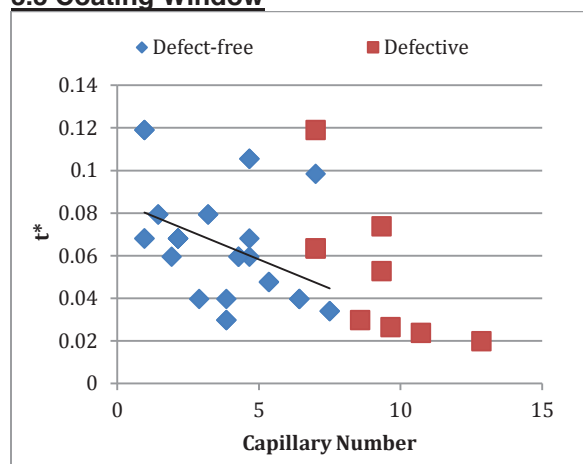


FIGURE 6. Coating process window for the installed slot-die coater. The y-axis denotes t^* , which is the ratio of expected minimum film thickness with the coating gap and the x-axis is the Capillary number given by $\mu V/\gamma$, where μ is the viscosity, V is the coating speed and γ is the surface tension. A trend of decreasing minimum film thickness with increasing speed is found.

The coating window for slot-die coating has been studied in the literature [3,4]. In general, it has been identified that a stable coating bead depends on three dimensionless numbers: the ratio of the far-field film thickness with the coating gap, the capillary number and the reynolds number. In addition to the stability of the coating bead, these numbers also determine the nature of the coating process, which can be broken down into three regimes. In the first regime, the minimum film thickness increases with speed. In the second regime, the minimum film thickness is more or less constant with speed and in the third, it decreases with coating speed. Clearly, the first regime is not desirable because it leads to a

trade-off between throughput, minimum film thickness and defect-free coating. The most desirable regime is the third because it allows pushing towards higher throughput while simultaneously getting lower film thicknesses. In experiments, this desirable trend was observed on flexible substrates before the coatings suffered from air entrainment and other defects related to high speed, as seen in Figure 6. In order to fully characterize this, more experiments need to be conducted with a wider variety of materials and process parameters.

3.4 Film Thickness

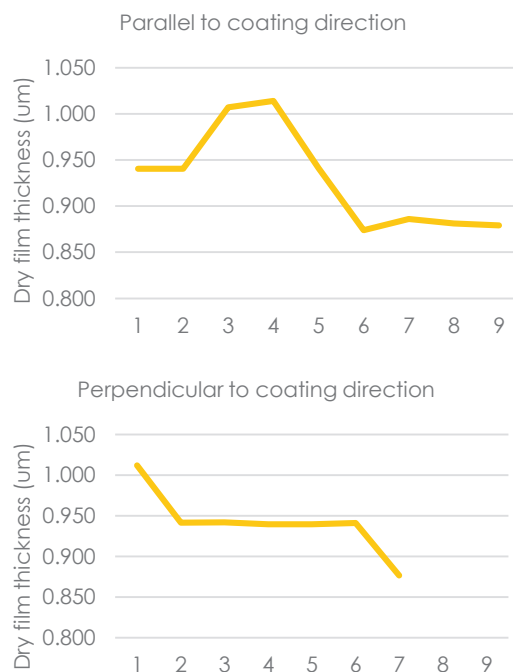


FIGURE 7. Measured cured film thickness along an arbitrarily chosen line in each, parallel and perpendicular to the coating direction on a test rigid substrate.

Preliminary experiments on the slot-die coater consisted of coating a photoresist solution with 20% solids content. This material was able to give O(1um) thick uniform films with a non-uniformity of ~10% on rigid substrates as seen in Figure 7. To drive film thickness down further, the material was diluted using PGMEA to ~2% solids content. With this formulation, it was possible to get ~40nm in film thickness at a coating speed of ~40mm/s on flexible substrates. However, this measurement was only obtained over a small area because of the lack of a large area metrology tool capable of measuring nanoscale films on flexible substrates. These issues are

being worked on in conjunction with further optimization of the slot-die coating process. If the 40nm thickness were to hold in its uniformity spec, it could potentially allow the realization of nanoscale lithography around it.

4. DISCUSSION

One of the most important considerations for nanoscale lithography is the ability to deposit nanoscale films at acceptable throughput. The two processes have been compared against each other in Table 1 for the metrics discussed in Section 1.3.

TABLE 1. Comparison of slot-die coating with inkjetting for R2R-UVIL for sub-100nm films.

Metric	Inkjetting	Slot-die Coating
Throughput	100mm/s	40mm/s
Minimum film thickness	<20nm	~40nm
Thickness uniformity	<5% ¹	No data to-date ²
Material compatibility	Low	High
Material wastage	Low	High

In general, the ability to deposit films of nanoscale thickness directly correlates with the ability to create the initial pattern and then also transfer the pattern onto an underlying substrate. This can be understood from an example. For a line/space pattern with 50% duty cycle and a height of 100nm, the minimum film thickness needed is about 65nm. This would result in an acceptable residual layer thickness of 15nm. Anything above 65nm would result in an undesirable, thick residual layer. The thicker the residual layer, the more losses in critical dimension when etching to transfer the pattern. Inkjetting has shown that it can produce the desirable range of film thicknesses to enable R2R-UVIL. While sub-100nm films have been potentially achieved for the slot-die coating tool, there are unanswered questions that are still in the process of being explored. The first question relates to the uniformity of the deposited film. The unavailability of a tool capable of measuring films on transparent flexible substrates hampered the uniformity characterization of the deposited films, as stated above in Section 3.4.

The other major concern relates to the fact that the obtained sub-100nm films were obtained after thermal curing for the photoresist formulation. The starting solution had ~2% solids content,

thereby implying that the wet films were > 1um in thickness. Hence, the UV imprint resist will need to be formulated with enough solvents that evaporate quickly and lead to a film of the desired imprint resist [5]. Also, the slot die coater, curing station, and the imprint module would all have to be placed in an integrated R2R tool to avoid the need for transporting wet films from the coater to the imprint tool. Given that slot-die coating can enable a broader variety of materials to be explored, including 1D and 2D nanomaterials-based resists, the challenges in machine design and achievable film thickness are worth pursuing.

REFERENCES

- [1] Sreenivasan S. V., Choi B. J., Schumaker, P. D., Xu, F. Status Of UV Imprint Lithography For Nanoscale Manufacturing. Handbook of Nanofabrication. Elsevier: 2011.
- [2] Derby B. Inkjet Printing of Functional and Structural Materials: Fluid Property Requirements, Feature Stability, and Resolution, Annual Review of Materials Research. 2010; 40:395-414.
- [3] Carvalho M. S., Kheshgi H. S. Low flow limit in slot-die coating: Theory and experiments, AIChE Journal. 2000; 46(10): 1907-1917.
- [4] Chang Y., Chang H., Lin C., Liu T., Wu P. Three minimum wet thickness regions of slot die coating, Journal of Colloid and Interface Science. 2007; 308(1): 222-230.
- [5] Liu W., Xu F., Fletcher E. B., Solvent-assisted layer formation for imprint lithography, US Patent Application EP20080779665. 2011.

ACKNOWLEDGEMENTS

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¹ This data was measured on silicon wafers to circumvent the problem of lack of sub-100nm film metrology on transparent substrates.

² For sub-100nm films, metrology is not available on transparent substrates.

ULTRAPRECISION HIGH SPEED CUTTING

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INTRODUCTION

Historically, ultraprecision machining has evolved along two lines: towards higher accuracy, and towards greater complexity, culminating in today's capability of creating non-rotationally symmetric optical surfaces with submicron tolerances [1]. However, an important aspect, which has largely been neglected, is productivity. The development of Fast Tool Servo and Slow Side Servo turning has been an important step towards higher productivity, yet applications are limited due to dynamic and geometrical restrictions. Diamond milling techniques are almost universal, but are associated with rather long machining times. High speed cutting would cut down machining time by an order of magnitude. In conventional machining, high speed cutting offers higher material removal rates, reduced cutting forces and increased surface quality, at the expense of tool life [2]. The research presented in this paper is devoted to the question whether high speed cutting is applicable in diamond machining. We have come to a very promising conclusion.

REDUCTION OF CUTTING FORCES

If the cutting force with respect to the volume removed per unit time could be reduced by high speed cutting, larger depths of cut could be realized which would relax the demands on pre-machining tolerances. From a theoretical point of view, the observation of an exponential decrease of the cutting force with increasing cutting speed at constant material removal rate would be a clear indication of a transition to adiabatic shearing.

That an exponential drop of the cutting force does indeed occur over a certain cutting speed interval has been demonstrated in face turning experiments with increasing cutting speed from zero at the center to several kilometers per minute at the outer diameter (Fig. 1), in which the material removal rate (MRR) was kept constant by decreasing the feed per revolution

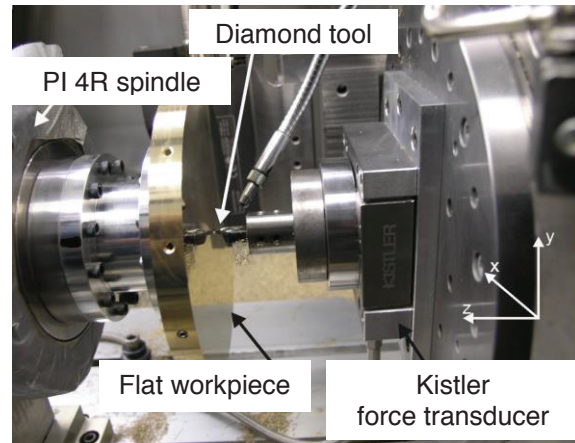


FIGURE 1. Face turning experiments with increasing cutting speed at constant material removal rate on a Nanotech 350 FG ultra-precision machine. Spindle speed: 9600 rpm. Workpiece diameter: 150 mm. Tool nose radius: 0.76 mm, 0° rake, 6° clearance.

inversely proportional to the distance from the center. We have extended the experiments reported earlier [3] to include OFHC copper and cartridge brass (CuZn30) whose flow behavior (i.e. dependence of the shear stresses on the shear rate) is known over a wide temperature range, which is a prerequisite for the modelling of high speed cutting processes. An FEM simulation of the cutting process would yield theoretical values of the cutting forces and estimates for stresses, strains and temperatures in the cutting zone.

Fig. 2 shows the cutting forces recorded with a Kistler 9256C1 force transducer in a face turning experiment with OFHC copper. In the speed interval between $v_c = 1$ km/min and 2 km/min an exponential decrease of the cutting force by about 50% is observed. This interval is shifted towards higher cutting speeds, if the depth of cut is reduced (while the feed is increased for maintaining a constant MRR) and to lower

speeds, if workpiece materials with greater hardness and higher modulus of elasticity are cut, which could be verified in face turning experiments with electroless nickel and brass alloys CuZn30 and CuZn39Pb3. The measured cutting forces were independent of the direction of cut and rather immune to the chip cross-section as long as the MRR was not changed.

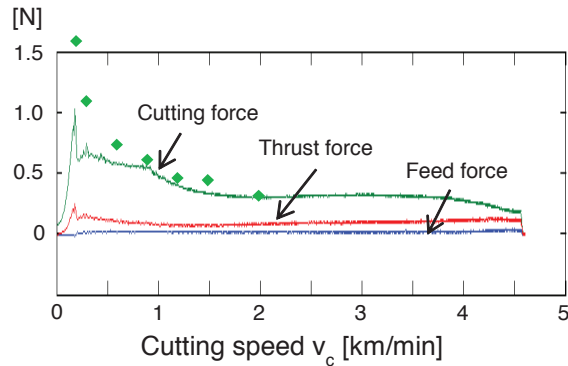


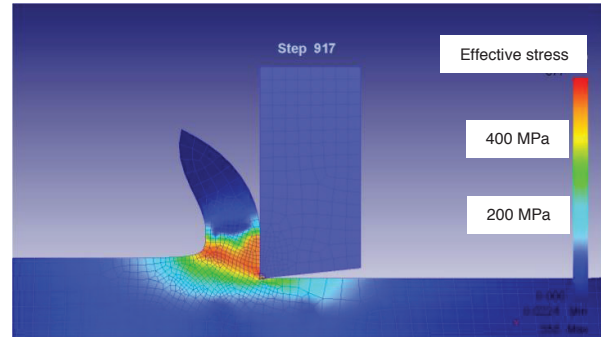
FIGURE 2. Measured cutting, thrust and feed forces and simulated cutting force (green diamonds) in face turning of OFHC copper. Depth of cut $a_p = 50 \mu\text{m}$. Material removal rate $\text{MRR} = 200 \text{ mm}^3/\text{min}$.

While 2D and 3D FEM simulations performed with DEFORM[®] with adaptive mesh yielded the correct order of magnitude and a roughly exponential decrease of the cutting force with increasing cutting speed, the details of the force variation observed in the experiments could not be reproduced (cf. Fig. 2). This should not be surprising, since the temperature dependent flow curves must be interpolated between known data points, the grain structure of the material is ignored, and guesses must be made regarding the friction coefficients between the chip surface and the rake and clearance faces of the diamond tool. Nevertheless, there is some confidence that the calculated stresses, strains and temperatures in the cutting zone represent the real situation. In particular, the simulation shows that the generated strains and heat are mainly confined to the chip at high cutting speeds and temperatures do rise by a factor of approx. 4 when the cutting speed increases from 0.15 km/min to 4 km/min.

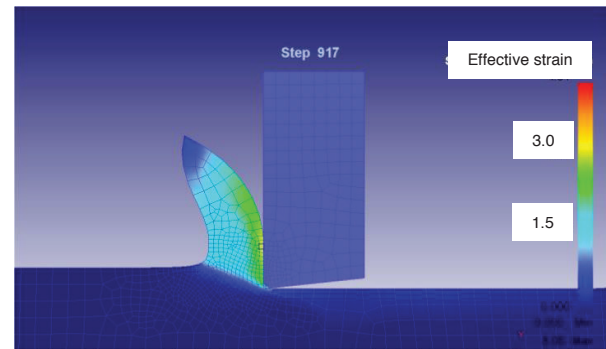
Both experiment and simulations prove that a transition to adiabatic shearing takes place and

cutting forces are largely reduced at high cutting speeds.

(a) Stress distribution



(b) Strain distribution



(c) Temperature distribution

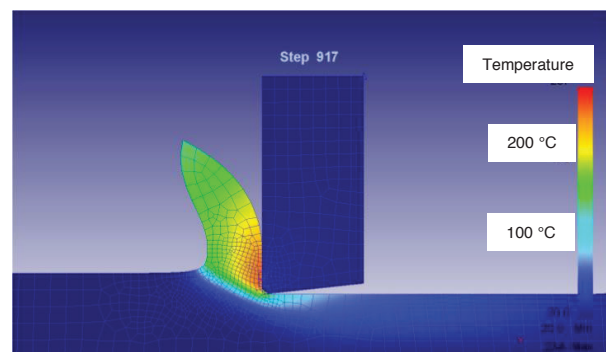


FIGURE 3. Distributions of the effective stress (a), effective strain (b) and temperatures (c) in the cutting zone obtained by 2D simulations of diamond turning of OFHC copper. Cutting speed $v_c = 4 \text{ km/min}$. Depth of cut $a_p = 10 \mu\text{m}$. Diamond tool: 0° rake, 6° clearance.

IMPROVEMENT OF SURFACE FINISH

The achievable roughness of a diamond turned metal surface is limited by the purity and the grain structure of the machined alloy. Since the elastic properties of a crystal are described by a tensor, and different grains have different relative orientations, the elastic recovery of the grains being cut in a diamond turning process differs from grain to grain. Hence, tiny steps appear at grain boundaries on the machined surface with step heights of a few nanometers. If cutting forces increase, e.g. when tool edges wear and get rounded, the step height between grains also increases (leading to so-called orange peel). If cutting forces decrease as in high speed cutting, does the step height between grains decrease as well?

Since the roughness of a diamond turned surface is generally dominated by the choice of machining parameters, material side flow, residual vibrations and straightness errors of the slides, experiments which are intended to trace delicate effects of the cutting speed on the amount of recovery (spring-back) of individual grains must be designed very carefully. We have devised two sets of experiments for evaluating the step height between the same grains being cut with a low cutting speed ($v_c = 0.1$ km/min) and then cut once again with a high cutting speed ($v_c = 2$ km/min). In the first set of experiments, isolated grooves were flycut with a radius tool, the second (fast) cut shifted vertically by $\Delta a_p = 5 \mu\text{m}$ (cf. Fig. 4). Selected grain boundaries were examined with an atomic force microscope after the first and after the second cut, and the step heights compared. In the second set of experiments, a V-shaped diamond tool with an included angle of 120° (which was inclined by 25° with respect to the normal of the surface) was used for flycutting of the grooves, the two successive cuts shifted horizontally by $\Delta a_e = 40 \mu\text{m}$ (cf. Fig. 5). Selected grain boundaries were scanned and the step heights on the two sides of the W-shaped profiles, cut at low and high speed, respectively, were compared. Thus it was possible to compare the step heights of the same grain boundaries cut at low and at high speed, admittedly not at the same spot (which is impossible) but at nearby spots either shifted vertically or horizontally.

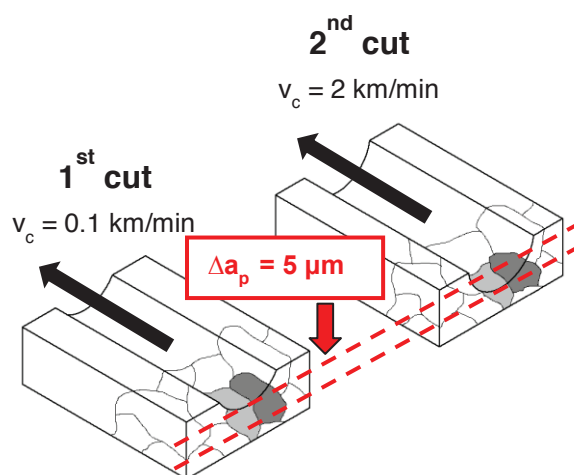


FIGURE 4. Profile milling with a radius tool (nose radius $r_s = 0.76$ mm). After evaluating the step heights between selected grains cut at low speed ($v_c = 0.1$ km/min), the groove is cut a second time at a high speed ($v_c = 2$ km/min) with a small vertical shift $\Delta a_p = 5 \mu\text{m}$.

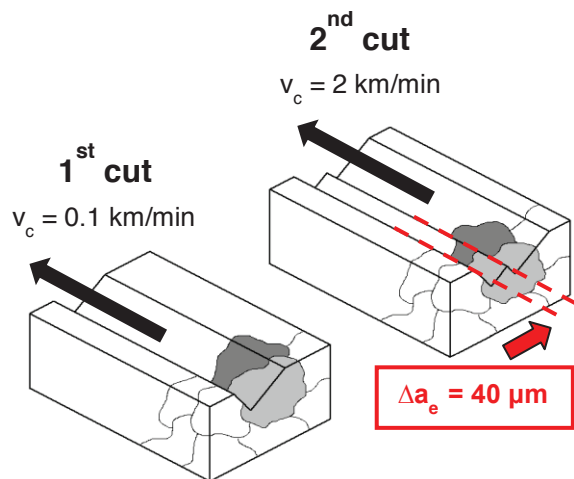


FIGURE 5. Profile milling with a V-shaped tool (included angle $\varepsilon = 120^\circ$). After evaluating the step heights between selected grains cut at low speed ($v_c = 0.1$ km/min), the groove is cut a second time at a high speed ($v_c = 2$ km/min) with a small horizontal shift $\Delta a_e = 40 \mu\text{m}$.

Altogether, 24 grain boundaries, 12 each in OFHC copper and brass CuZn39Pb3, which exhibited significant height differences larger than 10 nm when cut with a low speed, were selected for closer examination. Each step height was determined by averaging five profiles across the grain boundary. The histograms in Fig. 6 (copper) and Fig. 7 (brass) visualize the number of steps falling into a certain height interval, cut with both low and high speed. The histograms reveal that step heights tend to be smaller, if the material is cut with a high cutting speed.

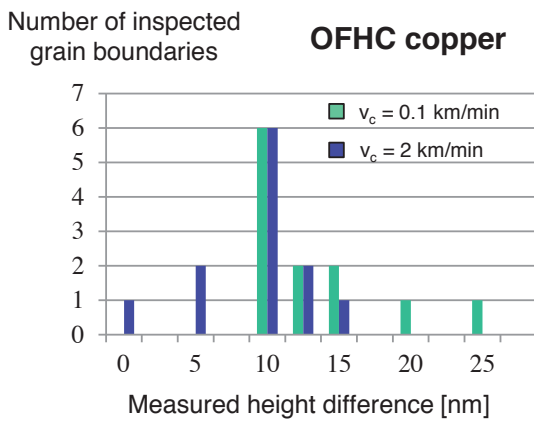


FIGURE 6. Histogram of step heights at grain boundaries in OFHC copper cut with low and high cutting speed.

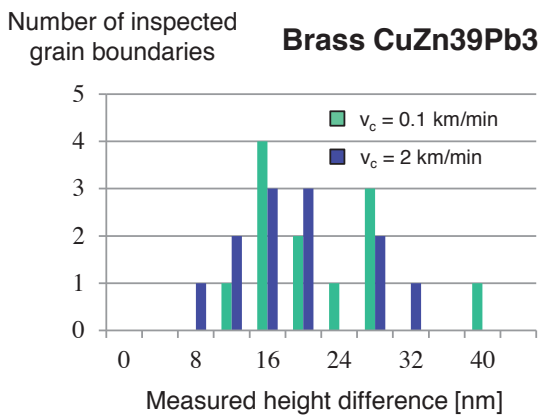


FIGURE 7. Histogram of step heights at grain boundaries in brass CuZn39Pb3 cut with low and high cutting speed.

REDUCTION OF DIAMOND TOOL WEAR

The crucial point in exploring the applicability of high speed cutting in diamond machining is tool wear. In experiments reported earlier [4] we have measured the flank wear rates of V-shaped diamond tools used in high speed cutting of electroless nickel and brass CuZn39Pb3 with a cutting speed $v_c = 4$ km/min and compared them with the wear rates obtained at a moderate cutting speed $v_c = 0.4$ km/min. The flank wear land FWL measured with an atomic force microscope (cf. Fig. 8) is plotted vs. cutting distance L_c , defined as the accumulated contact length per revolution, in Fig. 9.

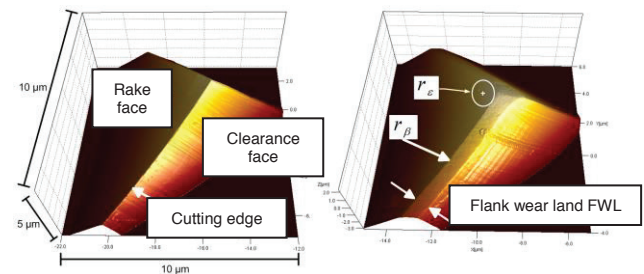


FIGURE 8. Atomic force microscope images of a new (left) and a worn diamond tool (right) used for flycutting of V-grooves with an included angle $\varepsilon = 90^\circ$.

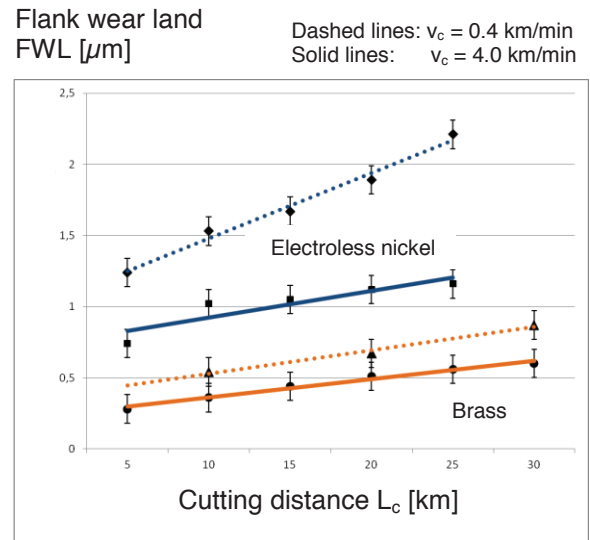


FIGURE 9. Measured flank wear land FWL vs. cutting distance L_c for electroless nickel and brass CuZn39Pb3.

In contrast to conventional machining, the wear rate of the diamond tool is reduced at high cutting speed. The reasons for this surprising fact could be the high thermal conductivity of diamond and the reduction of the cutting forces at high cutting speeds.

If abrasive wear is reduced in high speed diamond machining, what about chemically induced wear? Does the catastrophic tool wear observed in diamond machining of steel reduce, if the contact time per revolution in a high speed milling operation becomes comparable to the contact times in elliptical vibration cutting?

We have designed a milling experiment with well defined contact time in which a radius tool (nose radius $r_e = 0.76$ mm) is cutting through a thin C45 steel wedge with known width l_c (cf. Fig. 10) at a constant speed $v_c = 4$ km/min. By changing l_c , the contact time per revolution $t_c = l_c / v_c$ can be varied between $3 \mu\text{s}$ and $30 \mu\text{s}$.

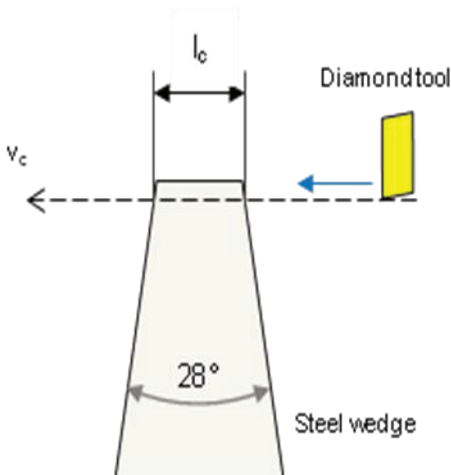


FIGURE 10. High speed diamond milling of a thin wedge with ($l_c = 0.2$ mm to 2 mm) with adjustable contact time per revolution t_c . The feed direction is parallel to the wedge.

In Fig. 11 the measured flank wear land FWL is plotted vs. cutting distance L_c for contact times per revolution $t_c = 3 \mu\text{s}$, $16 \mu\text{s}$ and $30 \mu\text{s}$ as well as for permanent contact (obtained from a turning experiment) which marks the rate of catastrophic tool wear. Obviously, the chemically induced wear depends on contact time and is largely reduced, if t_c is in the range of a few microseconds.

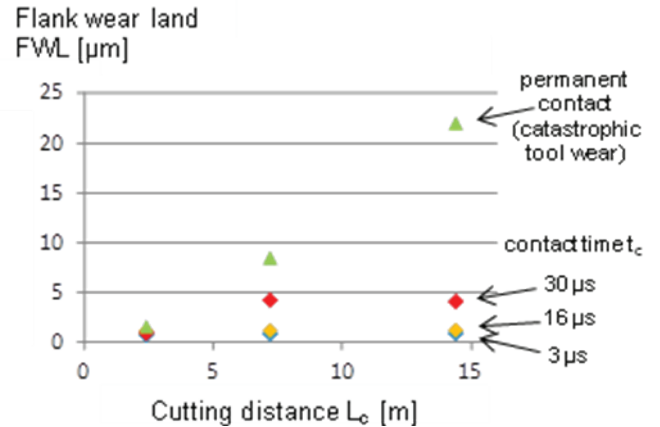


FIGURE 11. Flank wear land FWL vs. cutting distance L_c observed in diamond milling of C45 steel with different contact times per revolution t_c and for permanent contact.

For obtaining a reliable value of the flank wear rate at very short contact times, we have extended the milling experiment with $t_c = 3 \mu\text{s}$ up to a total cutting distance of 85 m (cf. Fig. 12). A comparison with Fig. 11 shows that chemically induced wear is reduced by a factor of ~ 30 compared to the catastrophic wear observed with permanent contact. Although this is not a breakthrough regarding practical applications (a factor of 1000 would be needed), the strong dependence on contact time might challenge theories which attribute the suppression of catastrophic wear observed in elliptical vibration cutting to the formation of an oxide layer [5].

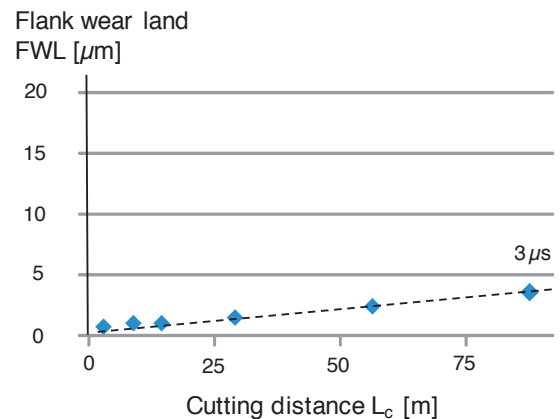


FIGURE 12. Flank wear land FWL vs. cutting distance L_c observed in diamond milling of C45 steel with a contact time per revolution $t_c = 3 \mu\text{s}$.

CONCLUSIONS

Our experimental and theoretical investigation of the high speed diamond machining of OFHC copper, brass alloys CuZn30 and CuZn39Pb3, electroless nickel and C45 steel leads us to the following conclusions:

- (1) A transition to adiabatic shearing takes place resulting in a reduction of the cutting forces.
- (2) The obtainable surface roughness does improve.
- (3) Abrasive as well as chemically induced tool wear is reduced.

These conclusions qualify diamond machining for high speed cutting applications with increased depth of cut and reduced machining times.

ACKNOWLEDGMENT

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REFERENCES

- [1] Brinksmeier E, Preuss, W Micromachining. Phil. Trans. R. Soc. A. 2012; 370: 3973-3992
- [2] Schulz H, Moriwaki, T High-speed machining. Annals of the CIRP. 1992; 41/2: 637-643
- [3] Brinksmeier E, Ohlsen I, Preuss W R Dependence of the cutting forces on cutting speed in high speed diamond turning. Proc. of the 28th Annual Meeting of the ASPE. 2014; 492-493
- [4] Brinksmeier E, Menten A, Preuss W R Tool wear in high speed diamond milling. Proc. of the 28th Annual Meeting of the ASPE. 2014; 494-495
- [5] Zhang X, Liu K, Kumar A S, Rahman M A study of the diamond tool wear suppression mechanism in vibration-assisted machining of steel. J Materials Processing Technology. 2014; 214: 496-506

CUTTING FORCE MEASUREMENT DURING DIAMOND FLYCUTTING OF CHALCOGENIDE GLASS

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INTRODUCTION

For ease of manufacture, optical devices have traditionally been based upon flat and spherical surfaces. Recent advances in manufacturing technology have made it possible to generate arbitrary freeform surfaces. The advantages include, for example: reduction of the number of components, reduction/elimination of some of the classical optical aberrations etc.

Ultraprecision diamond machining is a viable technology for the manufacture of freeform optics particularly for IR applications, where form tolerances and finish are less stringent than for visible applications. It has been known for several decades that brittle materials such as glass can be successfully diamond turned [1]. Such materials can be machined in a ductile mode if chip thicknesses are kept below a critical value [2-4]. However, studies on the machining behavior of IR glasses are limited; a better understanding of the cutting process is necessary to improve both component quality and process throughput.

In this paper, cutting force measurements during orthogonal turning and flycutting of arsenic selenium ($\text{As}_{40}\text{Se}_{60}$, trade name IRG26) chalcogenide glass are presented. This material has a glass transition temperature of 185°C , a refractive index of 2.82 at 2000 nm and >55% transmittance from 1100 nm to 12 μm . The high index and wide transmittance make this glass attractive for SWIR-LWIR applications. In addition to subtractive methods, the low transition temperature makes this glass viable for molding.

ORTHOGONAL TURNING EXPERIMENTS

Orthogonal machining experiments similar to those presented in [5] were conducted on a Moore Nanotechnology 350 FG machine with a 10,000 rpm main spindle in a temperature-controlled lab maintained at $20 \pm 0.1^\circ\text{C}$. Forces were measured while cutting the outer diameter of a 30 mm cylinder using a dead sharp (pointed) single crystal diamond (SCD) tool with 60°

included angle, 0° rake angle, and 10° clearance angle. This specially-designed tool consists of a 55° carbide DCMT-style insert to which the diamond is brazed, and a corresponding steel insert holder. The specimen was held with a spindle-mounted vacuum chuck which was finish-turned in situ to eliminate runout and resultant imbalance. The experimental arrangement and force directions are shown in Figure 1.

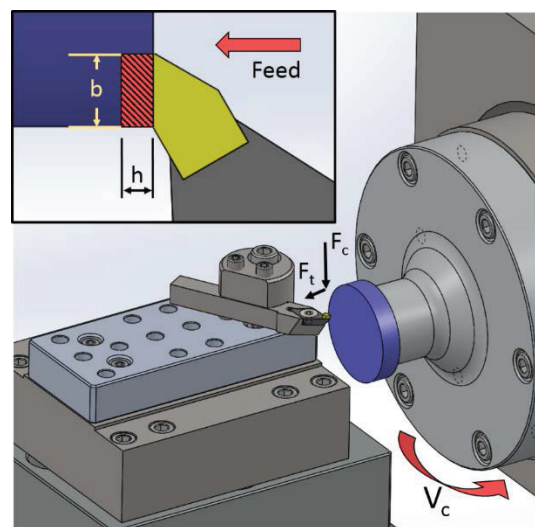


FIGURE 1. Experimental arrangement and cutting force orientation for orthogonal turning.

Cutting force (in the direction of cutting velocity) and thrust force (opposing the feed direction) were measured using a Kistler 9266C1 miniature cutting force dynamometer for a chip width (b) of 100 μm , a cutting speed (V_c) of 2 m/s, and chip thicknesses (h) from 200 nm to 2 μm . Four cutting tests were performed at each chip thickness. To reduce the effect of initial subsurface damage on subsequent tests, 250 revolutions of machining at a feedrate of 100 nm/rev were performed between each cutting experiment. To improve chip evacuation, tests were performed without lubrication, but with a gentle air blast on the tool tip. Data was acquired at 500 kHz using an NI USB-6251 device. Good agreement was seen between sets of experiments done in order of ascending and descending chip thickness.

Similar to the results presented in [5], cutting and thrust forces measured were much higher for h below 600 nm and increased linearly with h . While mechanisms are likely mixed, this is indicative of a ductile-dominated machining regime. Above this threshold, cutting forces drop sharply, suggesting a machining mode dominated by brittle fracture. Scatter among the experimental results is highest near this transition region. Results were found to be unchanged if orthogonal cutting was performed on the face rather than the periphery of the workpiece. Microscopy of tool edges before and after cutting showed little evidence of tool wear.

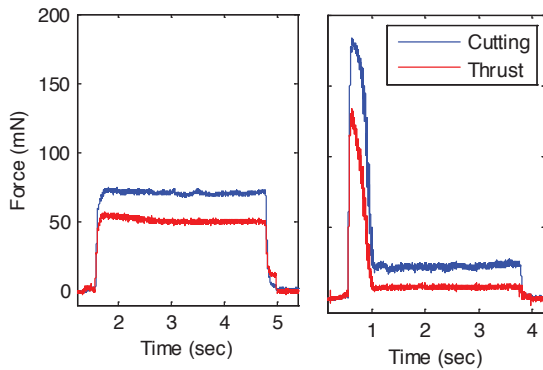


FIGURE 2. Orthogonal turning force signals at (left) $h = 400$ nm, (right) $h = 1.4$ μm .

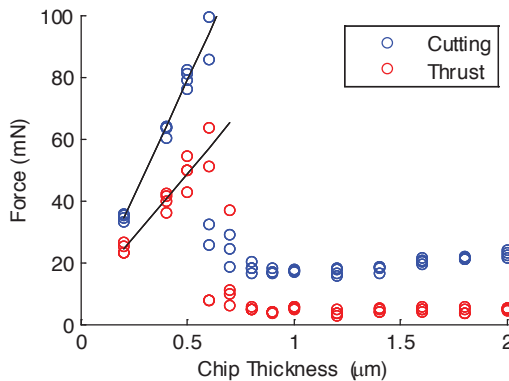


FIGURE 3. Cutting and thrust force variation with chip thickness, orthogonal turning.

Measured force signals for 80 revolutions of cutting are shown in Figure 2. In the ductile cutting regime, forces are constant throughout the cut. At higher chip thicknesses, the forces are larger at the beginning of the cut, and drop off quickly after several revolutions of cutting. This sharp force reduction after several revolutions is suggestive of forward-propagated damage, while the higher force levels at the beginning of the cut suggest successful removal of such damage between experiments.

Mean cutting and thrust forces, extracted from steady-state cutting behavior at each value of h are shown in Figure 3. Linear regressions over the ductile cutting regime indicate cutting coefficients of $K_t = 1490$ N/mm² and $K_c = 820$ N/mm².

SEM micrographs of resulting chips are presented in Figure 4. At a chip thickness of 400 nm, resulting chips are long and curly, similar to those seen when machining ductile metals. At higher chip thicknesses, chips are small and powder-like, suggesting cracking due to a fracture dominated process.



FIGURE 4. Chip morphology at 800X for (top) $h = 400$ nm and (bottom) $h = 1.4$ μm .

ORTHOGONAL FLYCUTTING EXPERIMENTS

While axisymmetric surfaces can be generated by diamond turning and freeform surfaces with sufficiently small departures can be produced using coordinated axis (slow- or fast-tool servo) turning, diamond milling can generate nearly arbitrary freeform surfaces useful for many optical applications. As described in [6], turning operations can be described as quasi-steady processes while interrupted milling operations often do not reach a steady state; there is evidence that this can lead to changes in the material removal mechanics. Several obstacles prohibit the direct comparison of cutting force data from simplified turning and milling operations: (1) milling cutters used in diamond

machining typically have large corner radii similar to a bullnose endmill, resulting in a more complex cutting geometry which is no longer orthogonal, and (2) at cutting speeds common in ultraprecision milling, frequency content of force signals exceeds the bandwidth of available transducers.

To enable an investigation of the mechanics of quasi-steady-state orthogonal turning and interrupted cutting operations, an orthogonal flycutting setup has been designed and constructed. The flycutter mounts to the turning spindle of the Moore Nanotechnology 350FG machine and holds the same cutting tool and insert used for the orthogonal turning experiments. The tool is mounted in a manner that preserves a 0° rake angle, 0° approach angle, and 10° clearance angle. With a flycutter radius of 50 mm, cutting speeds of 0.5 m/s to 8 m/s can be attained at spindle speeds between 190 rpm and 3050 rpm. The experimental arrangement and force directions are shown in Figures 5-6. This experiment preserves the orthogonal cutting geometry and allows an independent investigation of the effect of time of cut on the cutting mechanics.

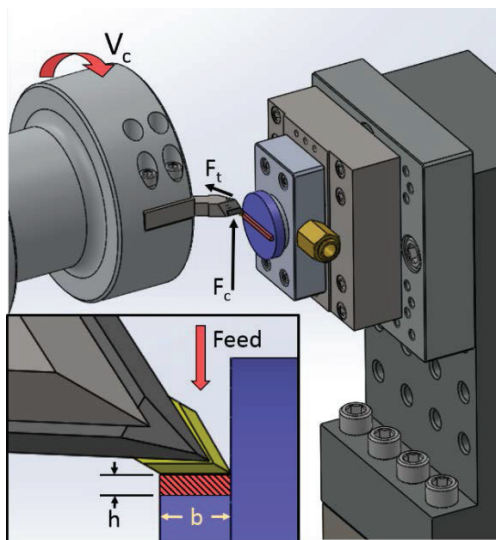


FIGURE 5. Experimental arrangement and force orientation for orthogonal flycutting.

For consistency, the same IRG26 sample was used as for the orthogonal cutting experiments. The sample was prepared using a round-nosed tool to produce a thin rib of material for flycutting tests. The rib-sample was mounted to a vacuum chuck which was attached to the cutting force dynamometer. With the centerlines of the cutter and rib coincident, orthogonal cutting conditions

comparable to turning were preserved. Though coupled, cutting speed and time in the cut could be varied by adjusting spindle speed and rib width.

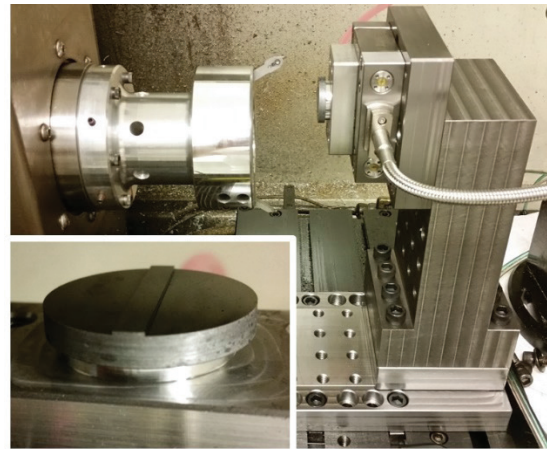


FIGURE 6. Experimental setup and specimen (inset) for orthogonal flycutting.

In this cutting configuration where the rib width is much smaller than the cutter radius, the force signals are impact-like in nature, exciting all the dynamic modes of the piezoelectric cutting force dynamometer. To remove these effects from the resultant force signal and extend the dynamic range of the measurement, inverse filtering [7] can be applied (see [8]).

Rib widths of 1, 2, and 4 mm, corresponding to tool-workpiece contact times of 500 μ s, 1.0 ms, and 2.0 ms were used. For each rib width, six experiments were performed at each chip thickness. As in orthogonal turning, surface speed was held to 2 m/s (350 RPM), while chip width was 100 μ m. Cleanup machining was performed at a feedrate of 100 nm/rev for 250 revolutions between each cutting experiment. No air blast or lubrication was necessary as chips were naturally ejected from the cutting zone. No differences were found between tests done in order of ascending or descending chip thickness.

Measured force signals from two experiments are shown in Figure 7. As in orthogonal turning, force levels are constant throughout the cut in the ductile regime, while cuts in the brittle regime begin with large forces and drop off rapidly.

After applying the inverse filter to remove the effects of dynamometer dynamics, the data was trimmed to include only the cutting data that has reached steady-state. Mean cutting and thrust forces are shown in Figure 8. A sharp decrease

in cutting and thrust forces occurs at a chip thickness between 800 nm and 1 μm indicating a change in cutting mechanics similar to orthogonal cutting. While the transition in cutting mechanics appears to occur at higher chip thickness in interrupted machining than in turning, no clear change in cutting force with time of contact is evident. Linear regressions performed in the ductile regime for the three fin widths indicate cutting coefficients of $K_c = 1200$ -1350 N/mm^2 and $K_t = 660$ -750 N/mm^2 .

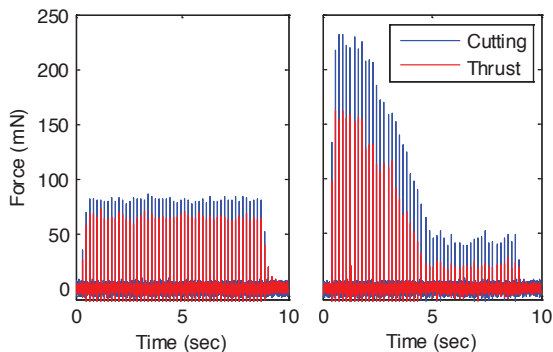


FIGURE 7. Orthogonal flycutting force signals at (left) $h = 400$ nm, (right) $h = 1.6$ μm .

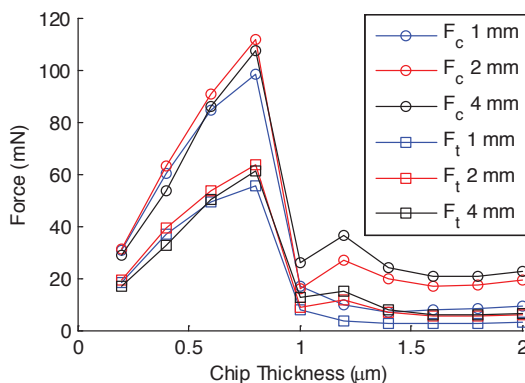


FIGURE 8. Cutting and thrust force variation with chip thickness, orthogonal flycutting.

REMARKS AND FUTURE WORK

Orthogonal turning and flycutting experiments were performed in $\text{As}_{40}\text{Se}_{60}$ (trade name IRG26) chalcogenide glass, which enable an initial comparison of quasi-steady and interrupted cutting processes. In both configurations, a transition from a high-force, ductile cutting behavior to a lower-force, brittle mechanism is present, as with other glasses [2-4]. In orthogonal turning, cutting coefficients similar to [5] were calculated, but a lower critical chip thickness of approximately 600 nm was observed. This variation may be due in part to variations in tool edge condition and material homogeneity. Lower cutting coefficients and a slightly higher critical

chip thickness between 800 nm and 1 μm were observed in flycutting experiments. In the brittle regime, a more gradual decrease in force was observed for flycutting compared to turning, possibly suggesting a change in crack propagation depth at shorter times of contact.

Future work will investigate lower times of contact (several μs), and higher cutting speeds (4-8 m/s), closer to those found in milling processes. Measurements will also be made to better understand the subsurface damage generated during machining operations.

ACKNOWLEDGEMENTS

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REFERENCES

1. Nakasui T, Kodera S, Hara S, Matsunaga H. Diamond Turning of Brittle Materials for Optical Components. CIRP Annals - Manufacturing Technology. 1990; 39(1): p. 89-92.
2. Blackley W, Scattergood R. Ductile-regime machining model for diamond turning of brittle materials. Precision Engineering. 1991; 13(2): p. 95-103.
3. Giovanola J, Finnie I. On the machining of glass. Journal of Materials Science. 1980; 15: p. 2508-2514.
4. Arif M, Rahman M, Yoke San W. Analytical model to determine the critical feed per edge for ductile-brittle transition in milling process of brittle materials. International Journal of Machine Tools and Manufacture. 2011; 51(3): p. 170-181.
5. Owen J, Davies M, Schmidt D, Urruti E. On the ultra-precision diamond machining of chalcogenide glass. CIRP Annals - Manufacturing Technology. 2015; 64(1): p. 113-116.
6. Schmitz T, Smith S. Machining Dynamics: Frequency Response to Improved Productivity New York: Springer; 2009.
7. Magnevall M, Lundblad M, Ahlin K, Broman G. High frequency measurements of cutting forces in milling by inverse filtering. Machine Science and Technology. 2012; 16(4): p. 487-500.
8. Ruben M, Schmitz T. Time domain modeling of compliant workpiece milling. In 30th ASPE Annual Meeting; 2015; Austin, TX.

DIAMOND TOOL WEAR WHEN MACHINING FERROUS METALS

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INTRODUCTION

Diamond turning (DT) can create a finished specular surface for optical components and precision optical molds. One application is the large drum molds used to transfer micrometer-scale features to plastic sheets in a process known as micro-replication. The drums must first be diamond turned before being made into high-quality optical molds for anti-reflective surfaces, laptop screens, infrared lenses for motion sensors and other optical components. As the applications for precision molds have expanded, wear-resistant materials such as steel have become more desirable than electroless Ni, hard copper or aluminum.

However, significant tool wear occurs during the DT of ferrous materials. The magnitude of the tool wear is 100x greater when machining steel compared to non-ferrous materials, such as aluminum. The high wear rate is believed to be the result of thermo-chemical effects, although the exact mechanisms are still unknown. Previous research [1-3] indicates towards two possibilities: diffusion of carbon into the workpiece or graphitization of the carbon atoms. Both of these are temperature dependent, but other factors could affect the rate of the tool wear.

Past work by Thornton and Wilks [3] has examined the combined effects of the surrounding atmosphere and machining parameters. At low speed ($<1\text{m/s}$), the effect of chemically reactive fresh material and oxygen saturation is believed to increase tool wear despite expected temperatures much lower than usually required for diffusion or graphitization [3]. At higher speed ($>11\text{m/s}$), a reduction in tool wear as a function of cutting speed was observed due to the inability of oxygen to saturate the material in the short contact time [3]. At these higher speeds, tool tip temperature is believed to be within the range of graphitization or diffusion and the oxygen merely acts as a catalyst. At all speeds, diamond wears at a much faster rate machining ferrous material due chemical interaction, but the exact mechanism is still unknown.

As the tool wears, the ability to produce a specular surface diminishes as well. Larger cutting edge radius, flank wear and the possibility of a Built-Up Edge (BUE) all lead to a degradation in the surface finish.

This paper describes a study of the parameters contributing to thermo-chemical diamond tool wear for cutting speeds from 1 to 6 m/s and examines the effect of speed on surface finish.

EXPERIMENTAL APPROACH

The experiments were designed to capture the dynamic cutting forces and tool temperature, while tracking changes in tool geometry. Experimental measurements provide insight into the relationship between machining parameters and tool wear rate. The presence of BUE on the tool and the shape of the resulting groove on the workpiece were also measured and compared to a theoretical model.

Two sets of experiments were conducted: the first with a straight edge tool cutting the outside diameter of a 0.8 mm thick disc and the second with a 0.5 mm radius diamond tool turning the face of a workpiece with a non-overlapping feedrate. The two diamond tools are shown in Figure 1.

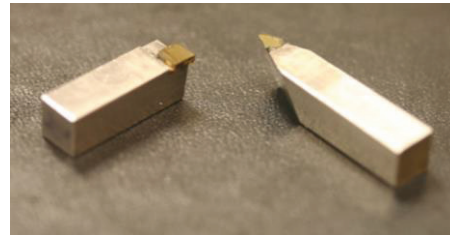


FIGURE 1. Two types of diamond tools were used during this research: a wide straight edge diamond tool (left) and a 0.5mm radius diamond tool (right).

CUTTING WITH A STRAIGHT-EDGE TOOL

AISI 1215 steel was cut using the straight edge diamond tool at 1, 2, 4 and 6 m/s for distances between 10 and 120m. No cutting oil was used and the Depth of Cut (DoC) was $2\mu\text{m}$. The diamonds were $4\times4\times1.5\text{ mm}$ in size with a clearance of 6° , a mass of .063 gr and were brazed to a tungsten alloy shank.

Temperature Measurements

A thin-film RTD was used to estimate the temperature of the diamond during machining. The RTD is in intimate contact with the back-side of the tool and responds quickly to a heat input. From the measurements at the back of the tool, the temperature at the cutting edge can be extrapolated using a finite difference heat transfer model. Figure 2 shows the estimated peak temperatures as a function of cutting speed.

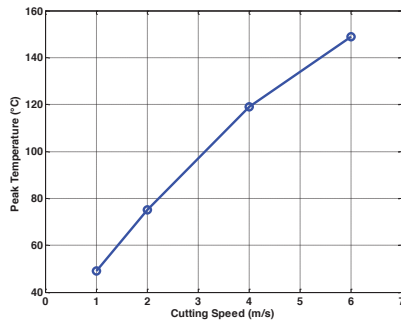


FIGURE 2. Estimated peak temperatures of turning 1215 steel at a $2\mu\text{m}$ DoC and 0.8mm wide piece.

Force Measurements

A three axis load cell is used to measure the cutting and thrust forces. Figure 3 shows the average forces as a function of cutting speed. Overall, forces increased slightly with cutting velocity. Potentially this could be due to BUE. As the speed increase, BUE becomes unstable and no longer contributes to the cutting geometry by the creation of a positive rake angle. Positive rake angles are known [4] to reduce cutting forces and this would account for lower forces at lower cutting speeds.

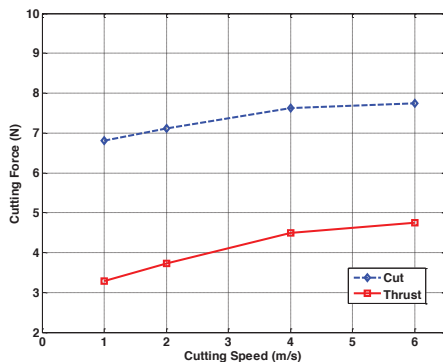


FIGURE 3. Force measurements at varying speeds in an orthogonal setup with a $2\mu\text{m}$ DoC of 1215 steel.

Wear Measurements

The SEM-EBID technique [2] developed at the Precision Engineering Center (PEC) was used to measure the changes to the tool edge after 10 m and 120 m of cutting distance for each cutting speed. Figure 4 shows the change in the tool edge cross section after 20m of cutting for the four cutting speeds. The new tool is also shown for comparison. As the speed increases, the cross sectional worn area increases believed to be due to the increase in tool tip temperature.

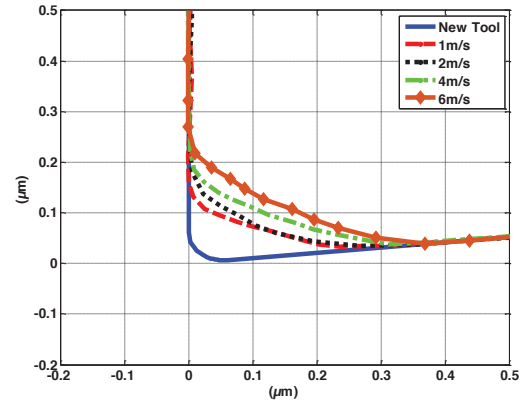


FIGURE 4. Wear measurements at 4 cutting speeds after 20m of cutting 1215 steel with a $2\mu\text{m}$ DoC.

The relationship between tool wear rate (defined as the change in cross section area per unit time - $\mu\text{m}^2/\text{s}$) and estimated peak temperatures was evaluated using the Arrhenius equation. The result was an estimate of the activation energy and pre-exponential constant given in Eq (1).

$$D = 167.7e^{\frac{-33.4}{RT}} \quad (1)$$

where D is the wear rate of diamond, is, R is the universal gas constant, and T is the absolute temperature at the tip of the tool. The activation energy and pre-exponential constant were found to be 33.4 kJ/mol and $167.7 \mu\text{m}^2/\text{s}$, respectively.

CUTTING WITH A ROUND-NOSE TOOL

Using a round nose tool, non-overlapping cuts were made at a $10\mu\text{m}$ DoC. Non-overlapping cuts produce complete tool grooves with symmetrical contact with respect to the nose of the tool.

Ten experiments were performed with a 0.5mm radius diamond tool: four with 1215 steel, four with 1045 steel and two with Aluminum 6061. Cutting speeds of 1 and 4 m/s were used at a cutting distance of 10m. 1215 steel and 1045 steel were both cut to compare the effect of the composition of the workpiece material. Al6061 was also cut with the same machining parameters to provide a reference for a diamond-turnable material when analyzing the tool grooves. Three experiments at 1 m/s were performed using 1215 steel and three were done at 4m/s using 1045 steel to determine the repeatability of the tool wear. Mobilemet 426 cutting oil was used and each surface was initially prepared with a carbide insert to have a surface finish better than $2\mu\text{m}$ peak-to-valley (PV).

Temperature Measurements

A thin film RTD was attached to the back of the tool. The diamond was approximately 2x2.5x0.75 mm with a 12° clearance and 0.5 mm radius.

At the same cutting speed, the RTD showed similar temperature measurements for the 1045 and 1215 steels. Using the same FDM model as in the previous section, peak temperatures were estimated. Only five measurements are shown in Figure 5 due to the chip interference on the sixth experiment. 1045 steel is expected to have higher peak temperatures due to higher shear strength.

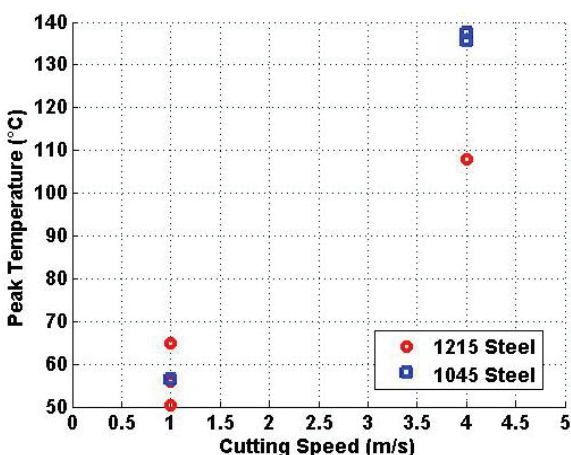


FIGURE 5. Estimated peak temperature for cutting 1045 and 1215 steel at 1 m/s and 4 m/s for 10m. Each was a 10 μ m DoC.

Force Measurements

The cutting and thrust force will change depending on the hardness of workpiece and the operating conditions. For a 10 μ m DoC at 1m/s, the average cutting and thrust force for 1045 steel were 7.8N and 3.8N respectively. Using the same machining conditions, the average cutting and thrust force for

1215 steel was 6.0N and 3.38N, respectively. This 26% difference in cutting force was expected due to 1045 steel being 20% harder. Hardness testing of each workpiece concluded 196 and 238 Vickers hardness for 1215 and 1045 steel, respectively.

Tool Wear Geometry

The tool wear on diamond tools can be characterized as flank wear and edge radius enlargement. The measured tool wear (from SEM and EBID) for the 0.5 mm tool radius was non-uniform over the contact zone. Figure 6 is a top down view of the wear land with the rake face of the tool on the bottom and the flank face at the top. Each line perpendicular to the cutting edge is from EBID. The width of the contact is 224 μ m wide. All of the wear occurs on the flank face and the increased wear at the edges is very noticeable. The cause of the large outer wear land may be due to higher availability of oxygen that acts as a catalyst [3]. At the outermost contact point on the tool, discontinuous cutting may occur due to the 2 μ m PV surface finish and allow more oxygen saturation.

The predicted wear land length for each experiment was 210 μ m using the relationship between the tool radius and depth of cut. Experimental results were slightly larger than the expected values. The discrepancy can be explained by the 2 μ m variation in the preliminary surface finish. A slightly varying DoC due to the carbide surface finish would explain the difference between the expected and actual lengths.

Effect of Cutting Velocity

Figure 7 shows the worn areas created while machining 1215 at 1m/s and 4m/s. The chip thickness (μ m) is overlaid on the graph with the vertical axis on the right side to compare with the cross sectional size of the wear.

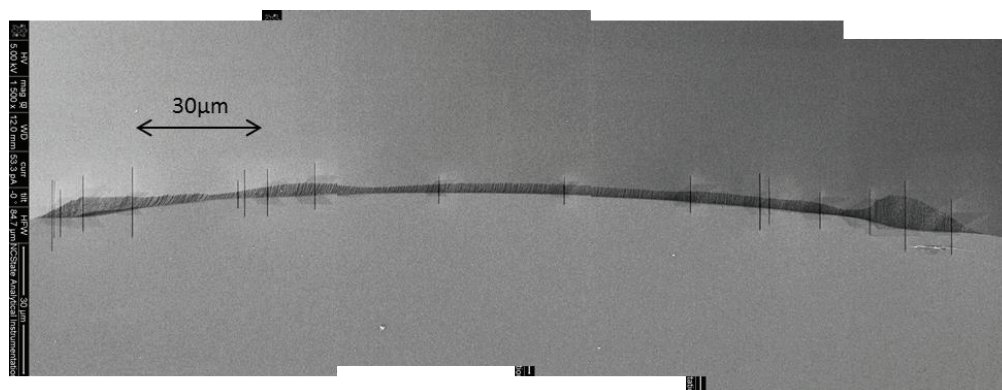


FIGURE 6. Wear of the tool edge after 10m of cutting 1045 steel at a 10 μ m depth and cutting speed of 4m/s. Vertical lines are from the SEM-EBID technique to quantify the wear of the tool edge.

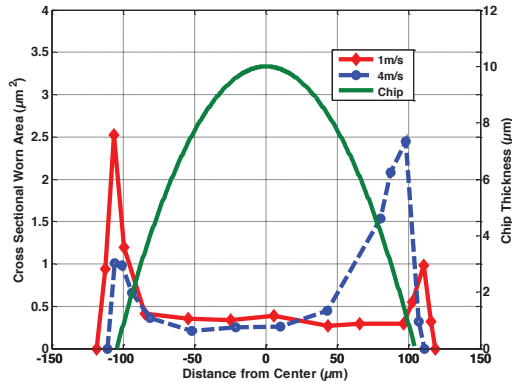


FIGURE 7. Magnitude of cross sectional worn areas after machining 1215 steel at a $10\mu\text{m}$ DoC for 10m at 1 and 4m/s.

A nonoverlapping cut creates a chip with symmetrical thickness about the center. Depending upon the DoC, the length and maximum thickness of the chip will change. The largest cross sectional worn areas were found on the edges of the wear land, where the chip thickness is minimal. Besides this localized areas of increased wear, the rest of the worn area is relatively constant.

Similar wear lands were measured for both speeds. Using these wear measurements, the total wear volume can be estimated as the area under the created line. In comparison, the 4m/s experiment wore $142\mu\text{m}^3$ of diamond, while the 1m/s experiment wore the $111\mu\text{m}^3$. This result is a 25% difference in comparison to a 39% difference that was found using the same machining parameters with a straight edge tool. 1045 steel did not exhibit a dependence on cutting speed with an actual 6% decrease in wear volume from 1 m/s to 4 m/s.

Effect of Workpiece Material

1045 steel has about four times the carbon content of 1215 steel. This increase in carbon was theorized to reduce wear by contributing to missing d-shell electrons by Paul and Evans [1]. Experimental results in Figures 8 and 9 contradict this assessment. Similar to Figure 7, the worn area measurements are graphed against with the chip thickness as comparison in Figure 8. 1045 steel accelerated wear at a much higher rate than 1215 steel. For 1045, the wear on the edges was five times higher than for 1215. Figure 9 shows the worn volumes for each steel experiment. The experiments that were done three times each gave results within 20% of the average.

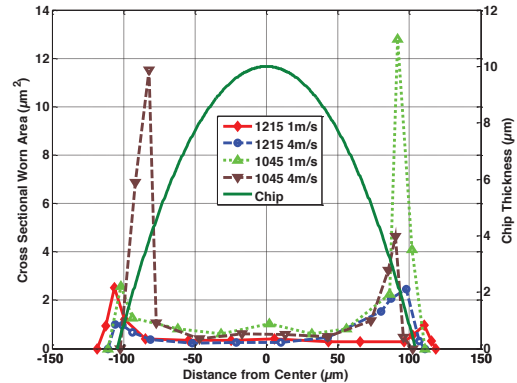


FIGURE 8. Magnitude of cross sectional worn areas after machining 1045 steel and 1215 steel at 1 and 4 m/s, $10\mu\text{m}$ DoC and 10m cutting distance.

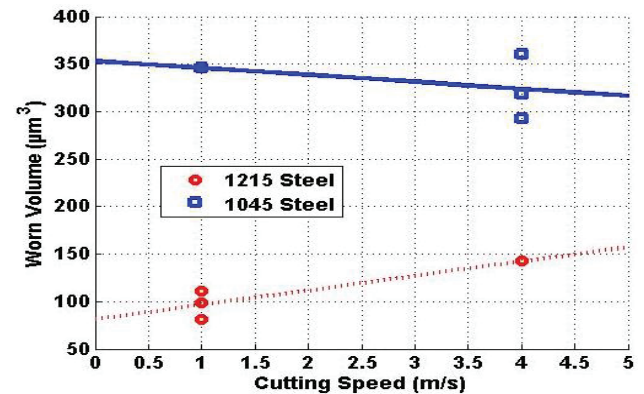


FIGURE 9. Estimated worn volume of diamond after 10m of turning 1215 and 1045 steel at 1 and 4 m/s.

Presence of Built-Up Edge (BUE)

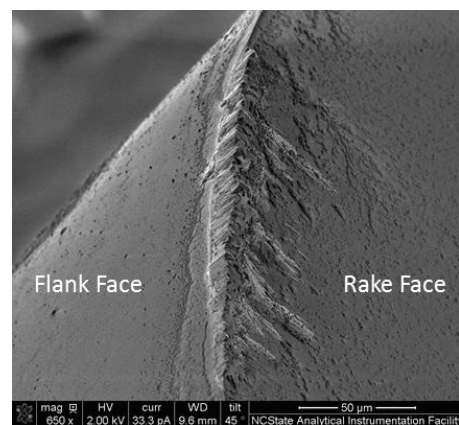


FIGURE 10. BUE after machining 1215 steel at 4m/s for 10m.

During machining, the presence of BUE occurs due accumulation of material on the tool's surface. After each experiment, the tool was

taken off and examined in the SEM while the BUE was still intact. Figure 10 shows the presence of a large amount of build on the rake face of the tool

Effect of BUE on Tool Geometry

Using the SEM-EBID method on a section of the BUE provides a look at the extension of the cutting edge from original tool. Figure 11 is cross sectional view of the BUE on the tool. BUE is considered a dynamic process with material rapidly building and falling off the tool once it becomes too large. The figure below shows that

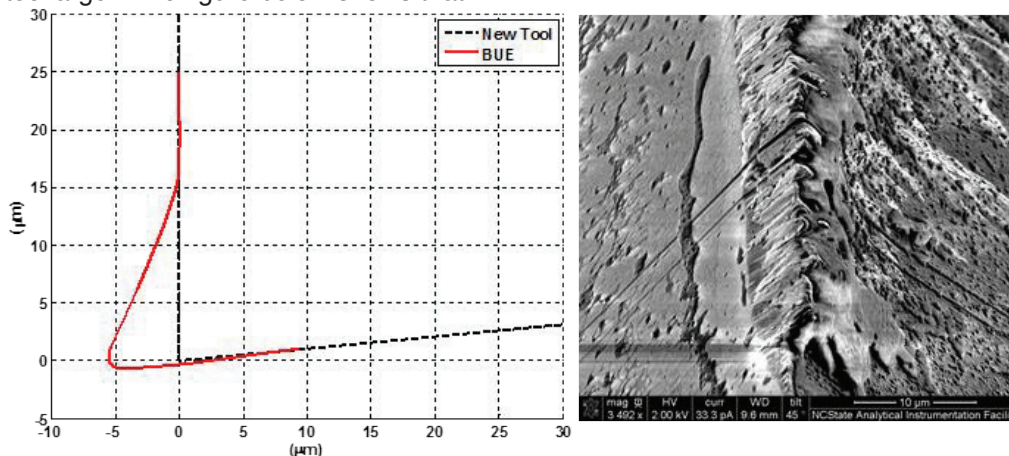


FIGURE 11. EBID measurement of BUE near the end of the tool-workpiece contact region.

Tool Groove Measurements

A Talysurf contacting profilometer was used to measure the grooves on the workpiece. Four measurements were made on each workpiece equally spaced at a 90° spacing (4 per rev). A MatLab program was used to compare the measured grooves with a theoretical tool model. Using the first groove cut with a new tool as a reference, the theoretical groove was located to minimize the area difference. This theoretical groove was then offset by the feedrate to compare the rest of the grooves.

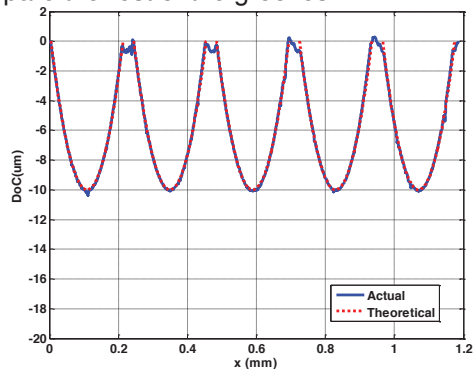


FIGURE 12. Tool grooves from machining Al6061 at 1 m/s.

BUE changes the tool geometry by increasing the DoC ($.5 \mu\text{m}$ in this case) and by enlarging the effective cutting radius (approximately $1.5 \mu\text{m}$). This enlargement reduces the repeatability of the groove left on the workpiece by the ever-changing tool geometry. By changing the cutting interaction from diamond-metal to metal-metal, the diamond is no longer susceptible to the fresh flowing ferrous material. This interaction may reduce tool wear by acting as a protective layer, but destroys the ability to give a high-specular surface finish.

Figure 12 and 13 show the comparison between the theoretical and actual grooves for machined Al6061 and 1215 steel workpieces. Al6061 showed little difference as the model lies directly on top of the results. 1215 steel changed more dramatically with noticeable jagged extensions in the middle of the tool grooves. This change in groove geometry is due to the BUE and tool wear. Observations showed Al6061 experiments had much less BUE when compared to 1215 or 1045 steel.

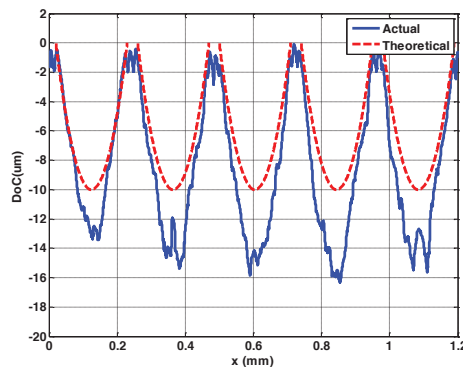


FIGURE 13. Tool grooves from machining 1215 steel at 1 m/s.

Away from the center of each groove the theoretical and actual converge quickly in the steel experiments; indicating that less build occurs as the localized chip thickness decreases. This is in agreement with Figure 10 where the buildup increases as it progresses toward the center of the tool.

Deterioration of Tool Groove

After analyzing one experiment of each material and each speed (6 total), averages of the area differential (between theoretical and actual tool grooves) for each groove were taken to present a trend line of deterioration as a function of cutting distance. Each trend line for 4 m/s is in Figure 14. Overall for 1215 steel, the tool grooves became progressively worse as a function of cutting distance. The 1045 steel did not produce this relationship, but consistently had grooves with larger area differences. For both materials, an increase in speed produced smaller area differentials from the theoretical. This is expected due to speed often being used to combat BUE [5]. In comparison to the Al6061 experiments, 1215 was a magnitude of order worse and 1045 was two magnitudes of order worse.

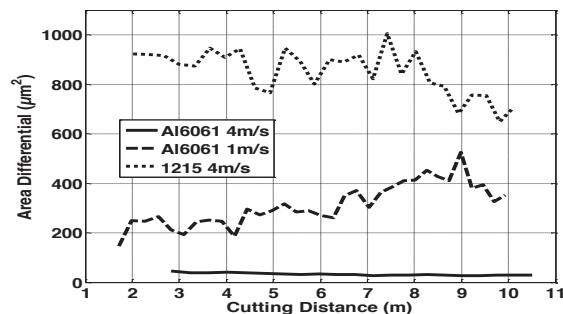


FIGURE 14. Comparison of tool groove deterioration against cutting distance for 1045, 1215 steel and Al6061 at 4m/s with a 10µm DoC.

CONCLUSIONS

Diamond turning can provide a specular surface and precise dimensional accuracy on many materials. Unfortunately, as the need to move toward harder materials, steel is not an option due to the accelerated chemical wear on the diamond. This tool wear is an extremely important parameter when dealing with submicron level surface finishes and shapes.

As expected [2,4], a relationship between temperature and wear rate was observed while machining 1215 steel with a straight edge tool. Wear increased as the speed was increased. Using a round nose tool and a non-overlapping cutting geometry, the wear was larger at each

edge of the chip. This result may support the theory of oxygen acting as a catalyst in the contact area more susceptible to atmospheric saturation (edge of contact). The chemical composition of the workpiece is believed to effect the tool wear rate, but not in the expected relationship. As the carbon content of the steel increases to potentially reduce the reactivity of the iron, the wear actually increased.

BUE was measured on the tool and is believed to be the main reason for the increased surface finish and groove deterioration. Initial experiments showed that increased cutting reduced the BUE and improved the finish. Potentially, this reduction in BUE could lead to higher tool wear rates by supplying more “fresh and chemically reactive” surfaces and higher tool temperatures. The combination of BUE, temperature and tool wear results in a dynamically changing interaction between the diamond tool and workpiece and makes it very difficult to model and optimize the diamond turning process.

REFERENCES

- [1] Paul E, Evans CJ, Mangamelli A, McGlaufflin ML, Polvani RS. Chemical aspects of tool wear in single point diamond turning. *Prec Eng.* 1996 1;18(1):4-19.
- [2] Lane BM, Dow TA, Scattergood R. Thermochemical wear model and worn tool shapes for single-crystal diamond tools cutting steel. *Wear.* 2013 3/15;300(1–2):216-24.
- [3] Thornton AG, Wilks J. Tool wear and solid state reactions during machining. *Wear.* 1979 3;53(1):165-87.
- [4] Hesler G. Investigating the Mechanisms of Diamond Tool Wear Cutting Ferrous Materials Using a Quantitative Study of Machining Parameters [MS Thesis]. North Carolina State University; 2014.
- [5] Shaw MC. *Metal Cutting Principles*. Crookall JR, Shaw MC, Suh NP, editors. New York, New York: Oxford University Press; 2005.

PROCESS DAMPING IDENTIFICATION FOR MULTIPLE DEGREE OF FREEDOM TURNING OPERATIONS

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INTRODUCTION

The analytical stability lobe diagram offers a predictive capability for selecting stable chip width-spindle speed combinations in machining operations. However, the increase in allowable chip width provided at spindle speeds near integer fractions of the system's dominant natural frequency is diminished substantially at low spindle speeds where the stability lobes are closely spaced. Fortunately, the process damping effect can serve to increase the chatter-free chip widths at these low speeds. This increased stability at low spindle speeds is particularly important for hard-to-machine materials that cannot take advantage of the higher speed stability zones due to prohibitive tool wear at high cutting speeds.

The objective of this study is expand on the work by Tyler and Schmitz [1] to identify and model process damping for multiple degree of freedom (MDOF) cutting systems. The level of process damping is measured for MDOF turning processes using flexure-based setups and compared to similar single degree of freedom (SDOF) results.

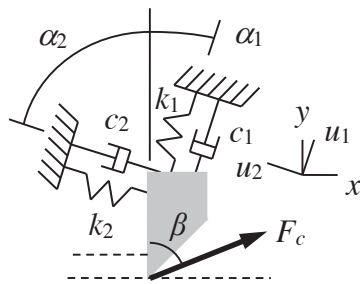


Figure 1: Turning model with a SDOF in two orthogonal directions.

MODEL

The turning model with a SDOF in two orthogonal directions is depicted in Fig. 1. The two mode directions, u_1 and u_2 , are oriented at the angles α_1 and α_2 , respectively, relative to the surface normal direction, y . The cutting force, F_c , is

oriented at the force angle β . The variable component of the cutting force is described by Eq. 1, where K_s is the specific cutting force coefficient, b is the commanded chip width, Y_0 is the vibration amplitude in the y direction from the previous revolution, and Y is the current vibration amplitude. The difference between Y_0 and Y identifies the variable chip thickness due to the vibration from one revolution to the next and provides the basis for regenerative chatter. The mean component of the cutting force is excluded because it does not influence stability for the linear analysis presented here.

$$F_c = K_s b (Y_0 - Y) \quad (1)$$

The assumption for Eq. 1 is that there is no phase shift between the variable force and the chip thickness. This is indicated by the real values of b and K_s . However, it has been shown that a phase shift can occur at low cutting speeds. This phenomenon is captured by the inclusion of the process damping force, F_d , defined in Eq. 2 [2], where C is the process damping coefficient, V is the cutting speed, and $\dot{y}$ is the tool velocity in the y direction. The process damping force is oriented in the y direction and opposes the cutting force (as projected in the y direction). In other words, it is a viscous damping force. Therefore, the process damping force is used to modify the structural damping and obtain an analytical stability solution.

$$F_d = -C \frac{b}{V} \dot{y} \quad (2)$$

To proceed with the solution, the cutting and process damping forces are projected into the u_1 and u_2 directions as shown in Eqs. 3-4, where F_{c1} and F_{c2} are the cutting force components in the u_1 and u_2 directions.

$$F_{u1} = F_c \cos(\beta - \alpha_1) - C \frac{b}{V} \dot{y} \cos(\alpha_1) = F_{c1} - C \frac{b}{V} \dot{y} \cos(\alpha_1) \quad (3)$$

$$F_{u_2} = F_c \cos(\beta + \alpha_2) - C \frac{b}{V} \dot{y} \cos(\alpha_2) = F_{c_2} - C \frac{b}{V} \dot{y} \cos(\alpha_2) \quad (4)$$

The time domain equations of motion for the two directions are provided in Eqs. 5-6, where m_i , c_i , and k_i , $i = 1, 2$, are the mass, viscous damping coefficient, and stiffness for the single DOF structural dynamics. In these equations, one overdot indicates one time derivative (velocity) and two overdots indicate two time derivatives (acceleration).

$$m_1 \ddot{u}_1 + c_1 \dot{u}_1 + k_1 u_1 = F_{c_1} - C \frac{b}{V} \dot{y} \cos(\alpha_1) \quad (5)$$

$$m_2 \ddot{u}_2 + c_2 \dot{u}_2 + k_2 u_2 = F_{c_2} - C \frac{b}{V} \dot{y} \cos(\alpha_2) \quad (6)$$

The y direction velocity can be written as a function of the velocities in the u_1 and u_2 directions as shown in Eq. 7. Substitution of Eq. 7 into Eqs. 5-6 yields Eqs. 8-9. Even though the structural dynamics are uncoupled (orthogonal), the equations of motion for the two directions are now coupled through the $\dot{u}_1$ and $\dot{u}_2$ velocity terms.

$$\dot{y} = \dot{u}_1 \cos(\alpha_1) + \dot{u}_2 \cos(\alpha_2) \quad (7)$$

$$m_1 \ddot{u}_1 + c_1 \dot{u}_1 + k_1 u_1 = F_{c_1} - C \frac{b}{V} (\dot{u}_1 \cos(\alpha_1) + \dot{u}_2 \cos(\alpha_2)) \cos(\alpha_1) \quad (8)$$

$$m_2 \ddot{u}_2 + c_2 \dot{u}_2 + k_2 u_2 = F_{c_2} - C \frac{b}{V} (\dot{u}_1 \cos(\alpha_1) + \dot{u}_2 \cos(\alpha_2)) \cos(\alpha_2) \quad (9)$$

By assuming a solution of the form $u_i(t) = U_i e^{i\omega t}$ for harmonic motion, Eqs. 8-9 can be rewritten in the frequency domain (ω is frequency). The results are provided in Eqs. 10-11, where the U_1 and U_2 terms have been grouped on the left hand side in both equations and the $e^{i\omega t}$ term has been dropped from both sides in each case.

$$\left(-m_1 \omega^2 + i\omega \left(c_1 + C \frac{b}{V} (\cos(\alpha_1))^2 \right) + k_1 \right) U_1 + i\omega \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right) U_2 = F_{c_1} \quad (10)$$

$$\left(-m_2 \omega^2 + i\omega \left(c_2 + C \frac{b}{V} (\cos(\alpha_2))^2 \right) + k_2 \right) U_2 + i\omega \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right) U_1 = F_{c_2} \quad (11)$$

These equations are arranged in matrix form as shown in Eq. 12, where:

- $a_{11} = \left(-m_1 \omega^2 + i\omega \left(c_1 + C \frac{b}{V} (\cos(\alpha_1))^2 \right) + k_1 \right)$
- $a_{12} = i\omega \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right)$
- $a_{21} = a_{12}$
- $a_{22} = \left(-m_2 \omega^2 + i\omega \left(c_2 + C \frac{b}{V} (\cos(\alpha_2))^2 \right) + k_2 \right)$

$$\begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} U_1 \\ U_2 \end{bmatrix} = \begin{bmatrix} F_{c_1} \\ F_{c_2} \end{bmatrix} \quad (12)$$

Using complex matrix inversion on a frequency-by-frequency basis, the direct and cross frequency response functions (FRFs) for the coupled dynamic system are obtained as shown in Eq. 13. The direct FRFs are located in the on-diagonal positions and the cross FRFs are located in the off-diagonal positions; the cross FRFs are equal because the inverted matrix is symmetric.

$$\begin{bmatrix} U_1 \\ U_2 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix}^{-1} \begin{bmatrix} F_{c_1} \\ F_{c_2} \end{bmatrix} = \begin{bmatrix} \frac{U_1}{F_{c_1}} & \frac{U_1}{F_{c_2}} \\ \frac{U_2}{F_{c_1}} & \frac{U_2}{F_{c_2}} \end{bmatrix} \begin{bmatrix} F_{c_1} \\ F_{c_2} \end{bmatrix} \quad (13)$$

Thusty [3] provided a frequency domain stability solution for regenerative chatter in turning, which defines the limiting stable chip width, b_{lim} , using Eq. 14, where $Re(G_{or})$ is the negative portion of the real part of the oriented FRF, G_{or} . This oriented FRF represents the projection of the cutting force into the mode direction and then the projection of this result in the surface normal direction.

$$b_{lim} = \frac{-1}{2K_s Re(G_{or})} \quad (14)$$

To relate the frequency-dependent b_{lim} vector to the spindle speed, Ω , Eq. 15 is applied to determine the relationship between and the valid chatter frequencies, f_c (i.e., those frequencies where $Re(G_{or})$ is negative). In this equation, $N = 0, 1, 2, \dots$ is the integer number of waves per revolution (i.e., the lobe number) and $\varepsilon = 2\pi - 2 \tan^{-1} \left(\frac{Re(G_{or})}{Im(G_{or})} \right)$ is the phase between the current vibration and the previous revolution.

$$\frac{f_c}{\Omega} = N + \frac{\varepsilon}{2\pi} \quad (15)$$

Thusty's approach is extended here to develop an oriented FRF that incorporates both the direct

and cross FRFs from Eq. 13. The oriented FRF is defined using Eq. 16, where μ_{ij} , $i, j = 1, 2$, are the directional orientation factors:

- $\mu_{11} = \cos(\beta - \alpha_1) \cos(\alpha_1)$ projects F into u_1 to cause u_1 vibration through the direct FRF $\frac{U_1}{F_{c1}}$ and then projects this result into y
- $\mu_{12} = \cos(\beta + \alpha_2) \cos(\alpha_1)$ projects F into u_2 to cause u_1 vibration through the cross FRF $\frac{U_1}{F_{c2}}$ and then projects this result into y
- $\mu_{21} = \cos(\beta - \alpha_1) \cos(\alpha_2)$ projects F into u_1 to cause u_2 vibration through the cross FRF $\frac{U_2}{F_{c1}}$ and then projects this result into y
- $\mu_{22} = \cos(\beta + \alpha_2) \cos(\alpha_2)$ projects F into u_2 to cause u_2 vibration through the direct FRF $\frac{U_2}{F_{c2}}$ and then projects this result into y .

$$G_{or} = \mu_{11} \frac{U_1}{F_{c1}} + \mu_{12} \frac{U_1}{F_{c2}} + \mu_{21} \frac{U_2}{F_{c1}} + \mu_{22} \frac{U_2}{F_{c2}} \quad (16)$$

The direct and cross FRFs included in Eq. 16 incorporate the process damping contribution by modifying the structural damping through the terms:

$$i\omega \left(c_1 + C \frac{b}{V} (\cos(\alpha_1))^2 \right), \quad i\omega \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right), \quad \text{and} \quad i\omega \left(c_2 + C \frac{b}{V} (\cos(\alpha_2))^2 \right)$$

as shown in Eq. 12. The process damping contribution depends on the $\frac{b}{V}$ ratio in each case, where $V = \frac{\pi d}{60} \Omega$ (d is the workpiece diameter and Ω is expressed in rpm). Therefore, the b and Ω vectors must be known in order to modify the damping. This establishes a converging stability solution. The following steps are completed for each lobe number, N :

1. the analytical stability boundary is calculated with no process damping to identify initial b and Ω vectors
2. these vectors are used to determine the initial process damping contribution
3. the stability analysis is repeated with the new damping terms to determine updated b and Ω vectors
4. the process is repeated until the stability boundary converges.

As shown in [4-6], the solution converges rapidly (20 iterations or less is typically sufficient).

The coupled dynamics solution is now extended to two DOF in the two orthogonal directions, u_1 and u_2 . From an FRF measurement in each direction, the modal parameters can be extracted (by peak picking, for example) which represent uncoupled single DOF systems in the modal

coordinates q_1 and q_2 for the u_1 direction and p_1 and p_2 for the u_2 direction [7]. This modal representation requires that proportional damping holds, but this is a reasonable approximation for the lightly damped tool dynamics typically observed in practice.

Equation 5, which provides the equation of motion for the u_1 direction with a single DOF, is rewritten in Eq. 17 to describe motion in the first modal DOF, q_1 . The y direction velocity is again $\dot{y} = \dot{u}_1 \cos(\alpha_1) + \dot{u}_2 \cos(\alpha_2)$, but $\dot{u}_1$ is now the sum of the modal velocities, $\dot{u}_1 = \dot{q}_1 + \dot{q}_2$; similarly, $\dot{u}_2 = \dot{p}_1 + \dot{p}_2$. Substitution yields Eq. 18. Equations 19-20 give the results for q_2 motion (the second modal DOF) in the u_1 direction.

$$m_{q1} \ddot{q}_1 + c_{q1} \dot{q}_1 + k_{q1} q_1 = F_{c1} - C \frac{b}{V} \dot{y} \cos(\alpha_1) \quad (17)$$

$$m_{q1} \ddot{q}_1 + c_{q1} \dot{q}_1 + k_{q1} q_1 = F_{c1} - C \frac{b}{V} \cos(\alpha_1) (\cos(\alpha_1) (\dot{q}_1 + \dot{q}_2) + \cos(\alpha_2) (\dot{p}_1 + \dot{p}_2)) \quad (18)$$

$$m_{q2} \ddot{q}_2 + c_{q2} \dot{q}_2 + k_{q2} q_2 = F_{c1} - C \frac{b}{V} \dot{y} \cos(\alpha_1) \quad (19)$$

$$m_{q2} \ddot{q}_2 + c_{q2} \dot{q}_2 + k_{q2} q_2 = F_{c1} - C \frac{b}{V} \cos(\alpha_1) (\cos(\alpha_1) (\dot{q}_1 + \dot{q}_2) + \cos(\alpha_2) (\dot{p}_1 + \dot{p}_2)) \quad (20)$$

Equations 18 and 20 are converted to the frequency domain by again assuming harmonic motion so that $q_j(t) = Q_j e^{i\omega t}$ and $p_j(t) = P_j e^{i\omega t}$, $j = 1, 2$. Equation 21 represents motion in Q_1 and Eq. 22 describes motion in Q_2 . Even though the modal degrees of freedom are uncoupled by definition, the two equations of motion for the u_1 direction now include both Q_1 and Q_2 due to process damping. Similar to the single DOF model in the previous section, the equations also include contributions from the u_2 direction dynamics (P_1 and P_2). Interestingly, the equations of motion are coupled in both modal coordinates and the two orthogonal directions. This presents a rich dynamic system which is unlike other machining models.

$$\begin{aligned} & (-m_{q1} \omega^2 + i\omega (c_{q1} + C \frac{b}{V} (\cos(\alpha_1))^2) + k_{q1}) Q_1 + \\ & \quad (i\omega C \frac{b}{V} (\cos(\alpha_1))^2) Q_2 + \\ & \quad (C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2)) (P_1 + P_2) = F_{c1} \end{aligned} \quad (21)$$

$$\begin{aligned} & \left(-m_{q_2} \omega^2 + i\omega \left(c_{q_2} + C \frac{b}{V} (\cos(\alpha_1))^2 \right) + k_{q_2} \right) Q_2 + \\ & \quad \left(i\omega C \frac{b}{V} (\cos(\alpha_1))^2 \right) Q_1 + \\ & \quad \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right) (P_1 + P_2) = F_{c_1} \end{aligned} \quad (22)$$

Following the same approach, the frequency domain equations for the u_2 direction are presented in Eqs. 23-24, where Eq. 23 describes motion in P_1 and Eq. 24 describes motion in P_2 .

$$\begin{aligned} & \left(-m_{p_1} \omega^2 + i\omega \left(c_{p_1} + C \frac{b}{V} (\cos(\alpha_2))^2 \right) + k_{p_1} \right) P_1 + \\ & \quad \left(i\omega C \frac{b}{V} (\cos(\alpha_2))^2 \right) P_2 + \\ & \quad \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right) (Q_1 + Q_2) = F_{c_2} \end{aligned} \quad (23)$$

$$\begin{aligned} & \left(-m_{p_2} \omega^2 + i\omega \left(c_{p_2} + C \frac{b}{V} (\cos(\alpha_2))^2 \right) + k_{p_2} \right) P_2 + \\ & \quad \left(i\omega C \frac{b}{V} (\cos(\alpha_2))^2 \right) P_1 + \\ & \quad \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right) (Q_1 + Q_2) = F_{c_2} \end{aligned} \quad (24)$$

Equations 21-24 are arranged in matrix form as shown in Eq. 25, where:

- $a_{11} = \left(-m_{q_1} \omega^2 + i\omega \left(c_{q_1} + C \frac{b}{V} (\cos(\alpha_1))^2 \right) + k_{q_1} \right)$
- $a_{12} = i\omega \left(C \frac{b}{V} (\cos(\alpha_1))^2 \right)$
- $a_{13} = i\omega \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right)$
- $a_{14} = i\omega \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right)$
- $a_{21} = a_{12}$
- $a_{22} = \left(-m_{q_2} \omega^2 + i\omega \left(c_{q_2} + C \frac{b}{V} (\cos(\alpha_1))^2 \right) + k_{q_2} \right)$
- $a_{23} = i\omega \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right)$
- $a_{24} = i\omega \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right)$
- $a_{31} = a_{13}$
- $a_{32} = a_{23}$
- $a_{33} = \left(-m_{p_1} \omega^2 + i\omega \left(c_{p_1} + C \frac{b}{V} (\cos(\alpha_2))^2 \right) + k_{p_1} \right)$
- $a_{34} = i\omega \left(C \frac{b}{V} \cos(\alpha_1) \cos(\alpha_2) \right)$
- $a_{41} = a_{14}$
- $a_{42} = a_{24}$
- $a_{43} = a_{34}$
- $a_{44} = \left(-m_{p_2} \omega^2 + i\omega \left(c_{p_2} + C \frac{b}{V} (\cos(\alpha_2))^2 \right) + k_{p_2} \right)$

$$\begin{bmatrix} a_{11} & a_{12} & a_{13} & a_{14} \\ a_{21} & a_{22} & a_{23} & a_{24} \\ a_{31} & a_{32} & a_{33} & a_{34} \\ a_{41} & a_{42} & a_{43} & a_{44} \end{bmatrix} \begin{bmatrix} Q_1 \\ Q_2 \\ P_1 \\ P_2 \end{bmatrix} = \begin{bmatrix} F_{c_1} \\ F_{c_1} \\ F_{c_2} \\ F_{c_2} \end{bmatrix} \quad (25)$$

Again using complex matrix inversion on a frequency-by-frequency basis, the direct and cross frequency response functions (FRFs) for the coupled dynamic system are obtained as shown in Eq. 26.

$$\begin{bmatrix} Q_1 \\ Q_2 \\ P_1 \\ P_2 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & a_{13} & a_{14} \\ a_{21} & a_{22} & a_{23} & a_{24} \\ a_{31} & a_{32} & a_{33} & a_{34} \\ a_{41} & a_{42} & a_{43} & a_{44} \end{bmatrix}^{-1} \begin{bmatrix} F_{c_1} \\ F_{c_1} \\ F_{c_2} \\ F_{c_2} \end{bmatrix} = \begin{bmatrix} \frac{Q_{1,1}}{F_{c_1}} & \frac{Q_{1,2}}{F_{c_1}} & \frac{Q_{1,3}}{F_{c_2}} & \frac{Q_{1,4}}{F_{c_2}} \\ \frac{Q_{2,1}}{F_{c_1}} & \frac{Q_{2,2}}{F_{c_1}} & \frac{Q_{2,3}}{F_{c_2}} & \frac{Q_{2,4}}{F_{c_2}} \\ \frac{P_{1,1}}{F_{c_1}} & \frac{P_{1,2}}{F_{c_1}} & \frac{P_{1,3}}{F_{c_2}} & \frac{P_{1,4}}{F_{c_2}} \\ \frac{P_{2,1}}{F_{c_1}} & \frac{P_{2,2}}{F_{c_1}} & \frac{P_{2,3}}{F_{c_2}} & \frac{P_{2,4}}{F_{c_2}} \end{bmatrix} \begin{bmatrix} F_{c_1} \\ F_{c_1} \\ F_{c_2} \\ F_{c_2} \end{bmatrix} \quad (26)$$

The direct FRFs in the u_1 and u_2 directions are defined by Eqs. 27-28, respectively; the cross FRFs are provided in Eqs. 29-30. The oriented FRF is again calculated using Eq. 16 and the directional orientation factors are the same.

$$\frac{U_1}{F_{c_1}} = \frac{Q_{1,1}}{F_{c_1}} + \frac{Q_{1,2}}{F_{c_1}} + \frac{Q_{2,1}}{F_{c_1}} + \frac{Q_{2,2}}{F_{c_1}} \quad (27)$$

$$\frac{U_2}{F_{c_2}} = \frac{P_{1,3}}{F_{c_2}} + \frac{P_{1,4}}{F_{c_2}} + \frac{P_{2,3}}{F_{c_2}} + \frac{P_{2,4}}{F_{c_2}} \quad (28)$$

$$\frac{U_2}{F_{c_1}} = \frac{P_{1,1}}{F_{c_1}} + \frac{P_{1,2}}{F_{c_1}} + \frac{P_{2,1}}{F_{c_1}} + \frac{P_{2,2}}{F_{c_1}} \quad (29)$$

$$\frac{U_1}{F_{c_2}} = \frac{Q_{1,3}}{F_{c_2}} + \frac{Q_{1,4}}{F_{c_2}} + \frac{Q_{2,3}}{F_{c_2}} + \frac{Q_{2,4}}{F_{c_2}} \quad (30)$$

The model may be extended to additional DOFs in each direction. For three DOFs in each direction, for example, Eq. 25 becomes a 6x6 symmetric matrix. The direct and cross FRFs are then a sum of six, rather than four, terms from the inverted matrix.

To demonstrate the multiple DOF algorithm, consider the model in Fig. 1 with $\alpha_1 = 30$ deg, $\alpha_2 = 60$ deg, $\beta = 70$ deg, $K_s = 2000$ N/mm², $C = 200$ N/mm, and $d = 75$ mm for an outer diameter turning operation. The structural dynamics are symmetric with a modal stiffness of 7×10^6 N/m, a natural frequency of 600 Hz, and a viscous modal

damping ratio of 0.03 (3%) for the first mode and a modal stiffness of 9×10^6 N/m, a natural frequency of 900 Hz, and a viscous modal damping ratio of 0.03 (3%) for the second mode. The corresponding stability limit with process damping effects is displayed in Fig. 2. To validate the predicted stability limit, the modal equations of motion were solved by Euler (numerical) integration in a time domain simulation [7]. Stable {spindle speed, chip width} combinations are identified by circles and unstable combinations are represented by squares in Fig. 2. Good agreement is observed.

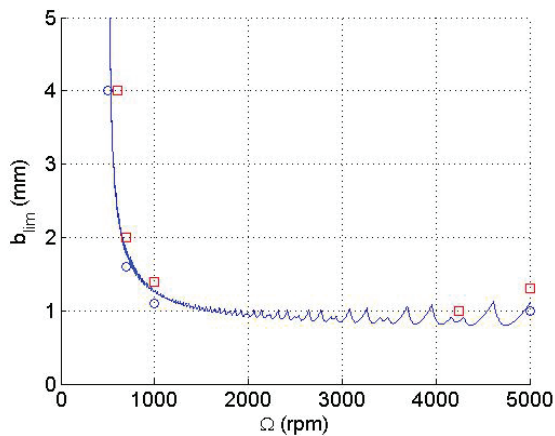


Figure 2: Comparison of analytical stability limit and time domain simulation results for turning model with 2DOF in two orthogonal directions. Stable {spindle speed, chip width} combinations are identified by circles and unstable combinations are represented by squares.

RESULTS

Low-speed cutting tests were performed in order to validate the multiple degree of freedom process damping model and calculate a process damping coefficient. The experiments were carried out on a Haas TL-1 CNC lathe. A custom notch hinge flexible cutting tool was designed to provide a multiple degree of freedom (DOF) system in the feed direction of the cutting operation; see Figure 3.

A tube turning geometry was selected for the cutting tests. In this arrangement, the feed direction is parallel to the tube axis. Because the flexure compliance was much higher than the workpiece in the feed direction, the stability analysis was completed using the flexure's dynamic properties only. The frequency response function and modal parameters for the flexure in

the feed and tangential directions are provided in Table 1.

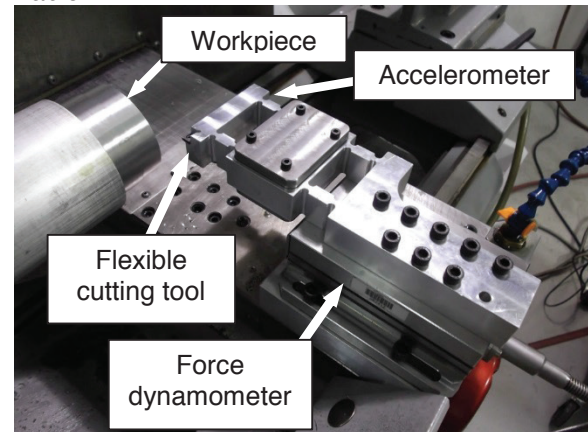


Figure 3: Experimental setup for turning stability tests. A cutting force dynamometer and accelerometer (not shown) were used to monitor the cutting process.

Table 1: Modal parameters for flexible cutting tool setup.

Direction	Viscous damping ratio (%)	Modal stiffness ($\times 10^7$ N/m)	Natur. freq. (Hz)
Tangential	7.21	3.71	703
Feed	6.23	0.80	303
	9.01	0.96	405
	1.68	1.88	1433

The workpiece material for all cutting tests was 6061-T6 aluminum. The wall thickness of the tube was varied, while a constant mean workpiece diameter of 87 mm was maintained for all experiments. The cutting insert (Kennametal SPEB322) was TiN coated with an 11 deg relief angle, 0 deg rake angle, and no chip breaker. The commanded chip width, b , varied between 1.0 mm and 3.0 mm and the cutting speed range was 82-273 m/min (300-1000 rpm). The feed rate was held constant at 0.13 mm/rev.

Before stability testing was completed, the force model for the workpiece/cutter combination was identified. This was achieved under stable cutting conditions using an off-the-shelf lathe insert holder that held the same insert as the manufactured flexible tool. The inserted holder was mounted to the cutting force dynamometer. The force model was calculated at both 500 and 1000 rpm for $b = 1$ mm and feed rate of 0.13 mm/rev. The mean values were $K_s = 1343 \pm 32$ N/mm² (which scales the chip area to obtain the resultant cutting force) and $\beta = 55^\circ$ (the force angle).

The stability was identified using an accelerometer (PCB Piezotronics model 352B10) and cutting force dynamometer (Kistler model 9257B). The force and vibration levels, as well as the frequency content of the two signals, were used to establish stable/unstable performance. Also, qualitatively, the machined surface finish after each test cut was inspected using a digital microscope.

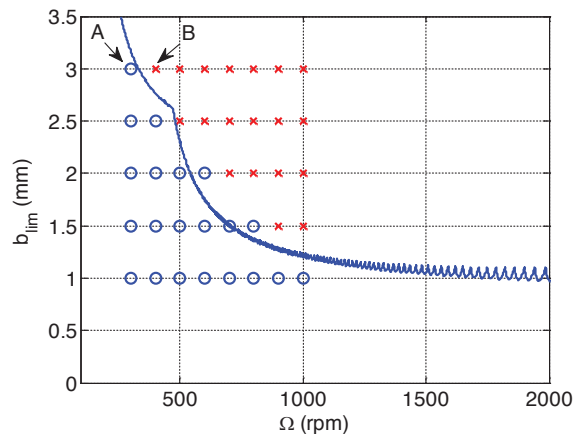


Figure 4: Stability boundary for the multi-degree of freedom system with the grid of stable (o) and unstable (x) cutting tests ($C = 1.3 \times 10^5$ N/m).

A grid of low-speed test points was selected to investigate the process damping behavior for the multi-degree of freedom system; see Figure 4. A process damping coefficient, C , was estimated based on a single variable residual sum of squares (RSS) minimization that best represented the stability boundary [6]. A process damping coefficient value of $C = 1.3 \times 10^5$ N/m was found to best represent the stability boundary. This results agrees with previously reported data for a SDOF setup [6].

CONCLUSIONS

An analytical stability model was presented which accounts for multiple degree of freedom vibratory systems in turning applications. The model includes a process damping force in the surface normal direction which depends on the depth of cut, the cutting speed, the tool velocity, and an empirical process damping coefficient.

REFERENCES

- [1] P.W. Wallace, C. Andrew, Machining forces: Some effects of tool vibration, *Journal of Mechanical Engineering Science*, 7 (1965) 152-162.

- [2] Y. Altintas, M. Eynian, H. Onozuka, Identification of dynamic cutting force coefficients and chatter stability with process damping, *Annals of the CIRP*, 57/1 (2008) 371-374.
- [3] J. Tlustý, M. Poláček, The stability of machine tools against self-excited vibrations in machining, in: *Proceedings of the ASME International Research in Production Engineering Conference*, Pittsburgh, PA, 1963, pp. 465-474.
- [4] C. Tyler, T. Schmitz, Process damping analytical stability analysis and validation, *Transactions of the NAMRI/SME*, 40 (2012).
- [5] C. Tyler, J. Karandikar, T. Schmitz, T., Process damping coefficient identification using Bayesian inference, *Transactions of the NAMRI/SME*, 41 (2013).
- [6] C. Tyler, T. Schmitz, Analytical process damping stability prediction, *Journal of Manufacturing Processes*, 15 (2013) 69-76.
- [7] T. Schmitz, S. Smith, *Machining Dynamics: Frequency Response to Improved Productivity*, Springer, New York, NY, 2009.

TIME DOMAIN MODELING OF COMPLIANT WORKPIECE MILLING

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INTRODUCTION

High performance application areas, such as the energy production, power, and aerospace industries, benefit from the enhanced product quality and reduced cost associated with machining thin-walled, metallic structures over the traditional fabrication and assembly methods. A methodology for machining compliant aluminum workpieces, described in [1-3], has been established and has become prevalent in the aerospace industry. The manufacturing strategy for these components consists of selectively removing material, via high-speed machining, from a solid billet to yield a monolithic component [4].

The superior mechanical properties of difficult-to-machine materials, such as titanium and nickel alloys, make them ideal candidates for compliant, thin-walled structures. However, the same machining methodology that has been applied to aluminum is not appropriate for these materials due to their excessive cost and limitations imposed by tool wear [5]. Near net shape techniques have been used to manufacture compliant structures composed of hard-to-machine materials, but these techniques are often unable to achieve the required dimensional tolerances. Because of the inherent compliance of the preforms, stable machining is difficult to achieve. Prediction of stable machining parameters is critical for the finish machining of such compliant workpieces.

The aim of this study is to evaluate the stability of milling operations where the workpiece is considerably more compliant than the machine-tool system. The evaluation is performed by implementing peak-to-peak (PTP) force diagrams as described in [6]. These diagrams result from multiple time domain simulations (TDS) completed over a range of spindle speeds and axial depths of cut. The outcome of an individual time domain simulation contains information specific to the spindle speed-axial depth of cut combination (i.e., cutting force, tool/workpiece deflection), while the PTP force diagrams contain

the global information provided by a stability lobe diagram. By proper choice of spindle speed and axial depth of cut according to the PTP force diagrams, stable machining parameters may be selected.

In this paper, we describe the time domain simulation model used in this study including the mechanistic force model and Eulerian integration approach to solving the dynamic equations of motion. The applicability of the simulation is validated through milling experiments where stability predictions are made based on pre-process knowledge of the tool/workpiece dynamics and cutting forces. Simulated preforms are machined in order to illustrate the process and show typical results. The conclusions summarize the usefulness of the simulation and highlight further work being conducted to improve the predictive capabilities of the model.

TIME DOMAIN SIMULATION MODEL

Based on the "Regenerative Force, Dynamic Deflection Model" described in [6-9], the time domain simulation determines the instantaneous chip thickness, calculates cutting forces, and uses Euler (fixed time step numerical) integration of the equations of motion to determine tool/workpiece deflections at each incremental time step. It is able to account for the nonlinearity which occurs during the milling process when the deflection of the tool/workpiece become large enough that contact is lost [10]. The underlying assumptions built into the model include the circular tooth path approximation and a mechanistic force model [11].

The instantaneous chip thickness depends on the feed per tooth, the relative vibration of the tool/workpiece in the surface normal direction for the current and previous cutting teeth, and cutter runout. Therefore, instantaneous chip thickness is expressed as:

$$h(t) = f_t \sin(\varphi) - n(t - \tau) - n(t) + r \quad (1)$$

where $f_t \sin(\varphi)$ is the nominal, tooth angle dependent chip thickness, $n(t - \tau)$ is the relative vibration of the tool/workpiece in the direction of the surface normal of the previous tooth, $n(t)$ is the current, relative vibration of the tool/workpiece in the normal direction, and r is the cutter runout. In these expressions, f_t is the feed per tooth, φ is the cutter rotation angle, t is the current time, and τ is the tooth passing period.

Cutting force calculations are based on the mechanistic force model presented by Budak et al. [11] and augmented with a process damping force [12]. At each incremental time step, the chip thickness is evaluated and, in the case where the tool has vibrated out of the cut (i.e., chip thickness is found to be less than or equal to zero), the instantaneous tangential, radial, and axial cutting forces are set to zero. For the case where the instantaneous chip thickness is non-zero, the instantaneous tangential, radial, and axial cutting forces can be expressed, respectively, as:

$$F_t^{i+1} = K_{tc} b h^{i+1} + K_{te} b - C_t b \frac{\dot{r}^i}{V} \quad (2)$$

$$F_r^{i+1} = K_{rc} b h^{i+1} + K_{re} b - C_r b \frac{\dot{r}^i}{V} \quad (3)$$

$$F_a^{i+1} = K_{ac} b h^{i+1} + K_{ae} b \quad (4)$$

where b is the axial depth of cut and h^{i+1} is the instantaneous chip thickness for the current time step. K_{tc} , K_{rc} , and K_{ac} are the tangential, radial, and axial specific cutting force coefficients, respectively, which are associated with “cutting” or shearing. The tangential, radial, and axial edge force coefficients, K_{te} , K_{re} , and K_{ae} , capture the ploughing effect which occurs at small chip thicknesses. The expressions $C_t b \frac{\dot{r}^i}{V}$ and $C_r b \frac{\dot{r}^i}{V}$ are the process damping forces in the tangential and radial directions, respectively, where C_t and C_r are the tangential and radial process damping coefficients, $\dot{r}^i$ is the velocity in the radial direction calculated in the previous time step, and V is the cutting speed. These instantaneous cutting forces are then transformed into the coordinate system shown in Figure 1.

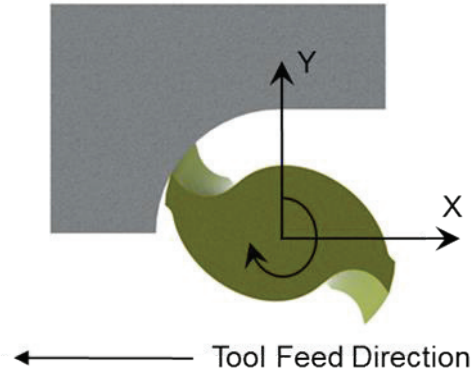


FIGURE 1. Coordinate system definition for the time domain simulation model. A down milling configuration is shown.

The equations of motion are solved in modal coordinates using Euler integration. The dynamics of the tool and workpiece are represented using modal parameters for an arbitrary number of degrees of freedom. The tool dynamics are considered in two orthogonal directions in the plane of the cut and the workpiece dynamics are considered in all three orthogonal directions. In modal coordinates the dynamic equation of motion may be expressed as:

$$F_q^i = m_q \ddot{q}^i + c_q \dot{q}^i + k_q q^i. \quad (5)$$

Then, as an approximated solution for velocity, $\dot{q}^{i+1}$, and displacement, q^{i+1} , via Euler integration:

$$\ddot{q}^{i+1} = \frac{(F_q^i - c_q \dot{q}^i - k_q q^i)}{m_q} \quad (6)$$

$$\dot{q}^{i+1} = \dot{q}^i + \ddot{q}^{i+1} \Delta t \quad (7)$$

$$q^{i+1} = q^i + \dot{q}^{i+1} \Delta t \quad (8)$$

where m_q , c_q , and k_q are the mass, damping, and stiffness values, respectively, expressed in modal coordinates, and Δt is the time step.

Additionally, the simulation model allows for a variety of tool geometries including an arbitrary number of cutting teeth, variable teeth spacing, different helix angles for each tooth, and cutter teeth runout. As a practical consideration it is

important to select a time step which is sufficiently small that the Euler integration method provides a numerically stable solution. A rule of thumb is that the time step should be at least ten times smaller than the period associated with the highest oscillation frequency present in the system being modeled. Also, the number of time steps (i.e., cutting tool revolutions) should be sufficiently high for the initial transient behavior to decay.

As previously mentioned, the outcome of individual time domain simulations contains information specific to the individual spindle speed-axial depth of cut combinations. This includes the instantaneous cutting forces and tool/workpiece deflections, velocities, and accelerations. The PTP force diagrams represent numerous time domain simulations performed over a range of spindle speed-axial depth of cut combinations. The range and step size of the spindle speed and axial depth of cut is specified, and the time domain simulation is performed for each combination. At the conclusion of each simulation the steady state portion of the time domain cutting forces are examined for the maximum peak-to-peak (PTP) force difference. The PTP force for each combination of spindle speed and axial depth of cut is used to generate a contour map over the range of spindle speeds and axial depth of cuts. The result is analogous to the traditional stability lobe diagram in the sense that it conveys a global representation of stable and unstable spindle speed and axial depth of cut combinations while retaining the specific, local information of the individual combinations.

EXPERIMENT DESCRIPTION

To validate the time domain simulation model, experiments were conducted. The cutting tests were performed on compliant workpieces, shown in Figure 2, which simulate near net shape preforms. These compliant workpieces were machined from 6061-T651 aluminum and are composed of two thin-walled structures with clamped-clamped-clamped-free (CCCF) boundary conditions. The geometric dimensions of the thin-walled structures, or ribs, are given in Table 1.



FIGURE 2. Model of the compliant workpiece.

The cutting tool used for the validation experiments was a 12.7 mm carbide end mill with two flutes, evenly spaced, and a 30° helix angle. Modal parameters for the tool were obtained via impact testing using an instrumented hammer (PCB 086C04) to provide the excitation force and a low mass accelerometer (PCB 352C23) to record the response. Table 2 lists the natural frequency, f_n , modal stiffness, k , and damping ratio, ζ , for the dominant modes in the plane of the cut (i.e., X-Y).

TABLE 1. Geometry of individual ribs.

Length	Height	Thickness
130 mm	15 mm	1.5 mm

Since cutting forces were used as the primary metric by which the simulation model was validated, the workpiece was bolted to a cutting force dynamometer (Kistler 9257B) during impact testing, as shown in Figure 3, to properly represent the workpiece dynamics during the validation tests. An instrumented hammer (PCB 084A17) was used to provide the excitation force and the workpiece response was measured using a non-contact laser vibrometer (Polytec OFV-534). Table 3 lists the natural frequency, f_n , modal stiffness, k , and damping ratio, ζ , for the two modes in the direction of greatest compliance (i.e., Y direction).

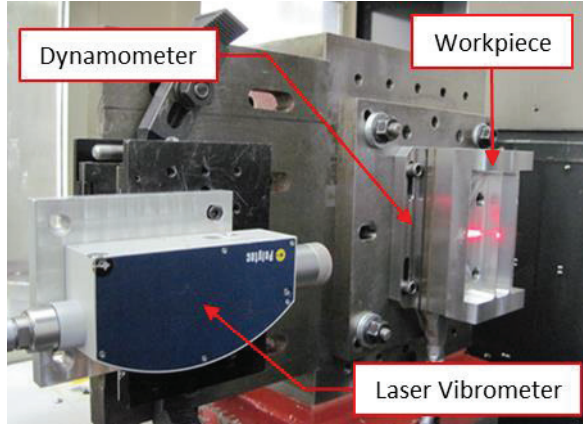


FIGURE 3. Impact testing setup for the compliant workpiece.

TABLE 2. Modal parameters for the cutting tool.

X direction		
f_n [Hz]	k [N/m]	ζ
2265	3.11×10^7	0.034
Y direction		
f_n [Hz]	k [N/m]	ζ
2254	3.09×10^7	0.027

The specific force coefficients used to calibrate the mechanistic force model applied in the time domain simulation were calculated using a constrained, least squares optimization method similar to the technique described in [13]. This method fits a set of simulated, instantaneous cutting forces to a set of measured, instantaneous cutting forces by iteratively updating the specific force coefficients until the convergence criteria is met. Table 4 lists the six specific force coefficients used in the simulation validation test.

TABLE 3. Modal parameters for the rib.

Rib 1			
Mode	f_n [Hz]	k [N/m]	ζ
1	5837	1.59×10^6	0.0007
2	7787	4.17×10^6	0.0008

TABLE 4. Specific force coefficients for the simulation validation test.

Cutting force coefficients		
k_{tc} [N/m ²]	k_{rc} [N/m ²]	k_{ac} [N/m ²]
1119	322	305
Edge force coefficients		
k_{te} [N/m]	k_{re} [N/m]	k_{ae} [N/m]
2	0	0

The simulations were performed at 2% radial immersion in a down milling configuration with a commanded feed rate of 0.1 mm/tooth. Flute-to-flute runout was specified as 3 μ m.

The PTP force diagram used in this validation study was generated over a spindle speed range from 7000 rpm to 9000 rpm at increments of 50 rpm, and the axial depth of cut ranged from 0 mm to 0.6 mm in increments of 0.1 mm. Computation time was approximately 5 minutes for the simulation. The PTP force diagram for the rib, shown in Figure 4, was used to select a spindle speed-axial depth of cut combination at which to perform a validation test. A stable cut, as predicted by the simulation, was chosen. A comparison of the simulated and measured, instantaneous cutting forces are provided in the following section.

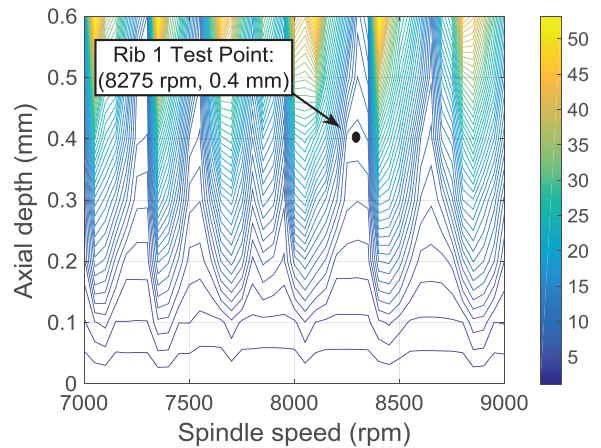


FIGURE 4. PTP force diagram indicating the selected, stable test point for rib 1 (the color bar indicates force in N).

EXPERIMENT RESULTS

The X direction cutting forces measured during the validation test, which was performed at a spindle speed of 8275 rpm and an axial depth of cut of 0.4 mm, are shown in Figure 5. It is observed that the measured and simulated cutting forces are in good agreement and the variation in the cutting force from one tooth engagement to the next, due to flute-to-flute runout, is also captured by the time domain simulation.

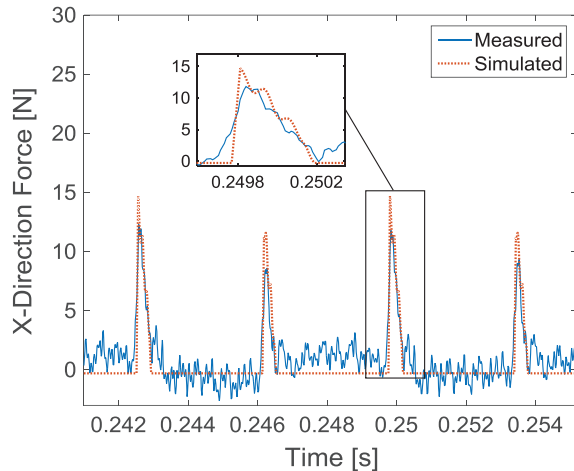


FIGURE 5. Measured and simulated X direction cutting forces shown for two cutter revolutions.

The frequency content of the X direction cutting force, computed using the Fast Fourier Transform (FFT), is shown in Figure 6. It is observed that the dominant frequencies are the tooth passing frequency, 276 Hz, and the runout frequency, 138 Hz, as well as their integer multiples (i.e., harmonics). Some amplification of the frequency content near the rib's natural frequencies is also evident.

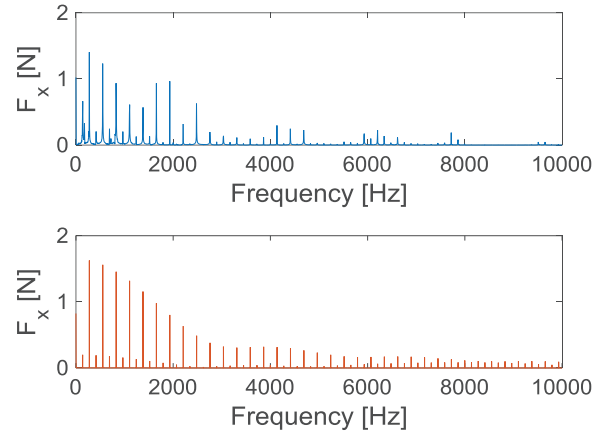


FIGURE 6. Frequency content of the measured (top) and simulated (bottom) X direction cutting forces.

The measured and simulated Y direction cutting forces, shown in Figure 7, also exhibit good agreement in both peak amplitude and frequency content. See Figure 8.

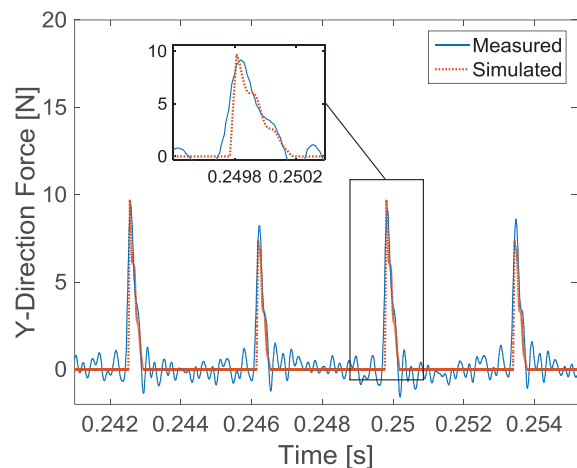


FIGURE 7. Measured and simulated Y direction cutting forces shown for two cutter revolutions.

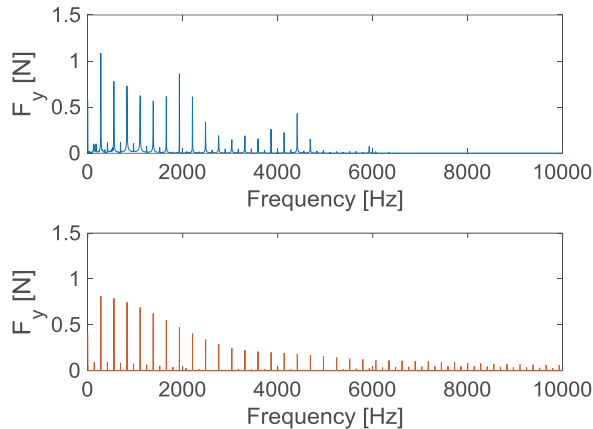


FIGURE 8. Frequency content of the measured (top) and simulated (bottom) Y direction cutting forces.

CONCLUSIONS

This paper presents the peak-to-peak force diagram and validates its applicability towards predicting stable and unstable conditions for compliant workpiece milling. The ability of the simulation to deliver the global stability predictions of traditional stability lobe diagrams has been demonstrated, and the specific, local information provided by the individual time domain simulations was validated.

For the purposes of milling compliant workpieces composed of hard-to-machine materials, the simulation model presented here may be augmented by including the process damping force. In the future, this work will be extended to include this force component to capture its stabilizing phenomenon at low cutting speeds.

REFERENCES

- [1] Tlustý J, Smith S, Winfough WR. Techniques for the Use of Long Slender End Mills in High-speed Milling. *CIRP Annals - Manufacturing Technology*. 1996; 45(1):393-6.
- [2] Smith S, Winfough WR, Halley J. The Effect of Tool Length on Stable Metal Removal Rate in High Speed Milling. *CIRP Annals - Manufacturing Technology*. 1998; 47(1):307-10.
- [3] Smith S, Dvorak D. Tool path strategies for high speed milling aluminum workpieces with thin webs. *Mechatronics*. 1998; 8:291-300.
- [4] Halley J, Helvey A, Smith K, Winfough W, editors. The impact of high-speed machining on the design and fabrication of aircraft components. *Proceedings of the 17th Biennial Conference on Mechanical Vibration and Noise*, 1999 ASME Design and Technical Conferences, Las Vegas, Nevada, September 12-16, 1999.
- [5] Smith S, Wilhelm R, Dutterer B, Cherukuri H, Goel G. Sacrificial structure preforms for thin part machining. *CIRP Annals - Manufacturing Technology*. 2012; 61(1):379-82.
- [6] Smith S, Tlustý J. Efficient Simulation Programs for Chatter in Milling. *CIRP Annals - Manufacturing Technology*. 1993; 42(1):463-6.
- [7] Schmitz TL, Smith KS. *Machining dynamics: frequency response to improved productivity*: Springer Science & Business Media; 2008.
- [8] Smith S, Tlustý J. An overview of modeling and simulation of the milling process. *Journal of engineering for industry*. 1991; 113(2):169-75.
- [9] Tlustý J. *Manufacturing processes and equipment*: Prentice Hall; 2000.
- [10] Tlustý J, Ismail F. Basic Non-Linearity in Machining Chatter. *CIRP Annals - Manufacturing Technology*. 1981; 30(1):299-304.
- [11] Budak E, Altintas Y, Armarego E. Prediction of milling force coefficients from orthogonal cutting data. *Journal of Manufacturing Science and Engineering*. 1996; 118(2):216-24.
- [12] Tyler CT, Schmitz TL. Analytical process damping stability prediction. *Journal of Manufacturing Processes*. 2013; 15(1):69-76.
- [13] Gonzalo O, Beristain J, Jauregi H, Sanz C. A method for the identification of the specific force coefficients for mechanistic milling simulation. *International Journal of Machine Tools and Manufacture*. 2010; 50(9):765-74.

PARALLEL LASER MACHINING AND METAL SINTERING BASED ON TEMPORAL FOCUSING

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In this paper, we present a 3-D resolved parallel laser micro-machining method based on ultrashort pulse lasers and temporal focusing, achieving 500 nm resolution over a $50 \times 50 \mu\text{m}^2$ processing area. This new method substantially improves the throughput of ultrashort pulse laser micro-machining which is limited by the sequential point-scanning processes. In the second part, we adapt the unique parallel laser processing technique to study metal sintering and melting process. The results indicate temporal focusing technique produces sintered/melted metal patterns with $30\sim 50 \mu\text{m}$ resolution over a $420 \times 260 \mu\text{m}^2$ in a parallel fashion, presenting significant advantages over conventional point-scanning based metal additive processes.

INTRODUCTION OF TEMPORAL FOCUSING

The concept of temporal focusing was first used in light microscopy in 2005 [1]. This optical design enables a two-photon microscope to obtain 2-D high frame-rate images, e.g. 200 fps, without using conventional mechanical scanning

components. In a temporal focusing system, the spectrum of a femtosecond laser pulse is first spatially separated by a diffraction grating, a lens then collimates these beams, and lastly the objective lens recombines the beams to the focal region. The laser pulse is broadened after the diffraction grating due to spectral-temporal relationship. Temporal focusing only occurs at the focal region because the different frequency components only spatially overlap within the focal region of the objective lens; thus the pulse width is the shortest only at the focal plane, enabling depth discrimination. At the focal region, the beams will not form a single spot as in conventional laser-scanning microscopes. Instead, due to the fact that all beams are spatially and temporally overlapped, the focal region becomes a “light sheet” with a diameter around a few hundred micrometers.

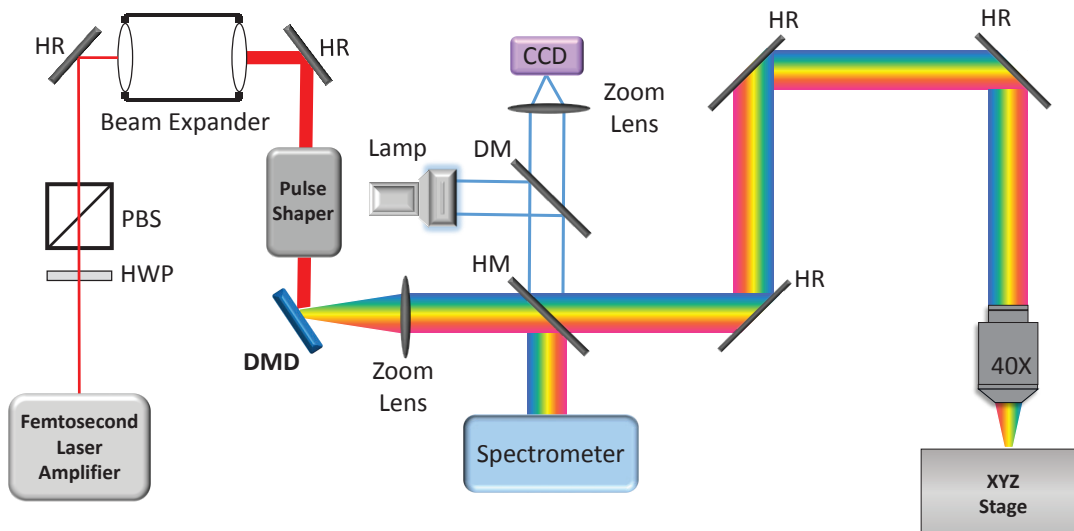


FIGURE 1. Schematic of the temporal focusing laser machining system (TFLMS); DM = Dichroic mirror; DMD = Digital micro-mirror device; CCD = Charge Coupled Device; HWP = Half wave plate; HR = High reflectance mirror; PBS = Polarized beam splitter; HM = Half mirror

PARALLEL LASER MACHINING

To realize 3-D resolved parallel laser machining, we combine the concept of temporal focusing with a digital micromirror device (DMD), where the DMD is simultaneously used as both a diffractive optical element and a programmable photomask. By controlling the DMD pixels and incident angle, one can form arbitrary patterns with different grating groove frequency, e.g. 600-1200 lines/mm.

The optical configuration of the temporal focusing laser machining system (TFLMS) is presented in Figure 1. In this system, the laser source is a Ti-Sapphire regenerative amplifier (Spectra-Physics, Spitfire Pro) with a center wavelength of 800 nm. The maximum pulse repetition rate of the laser amplifier is 1 kHz with a pulse duration of 100 fs. A half wave plate (HWP) and a polarized beam splitter (PBS) are used to control the laser power. The output laser beam is first sent to the beam expander, and guided to the DMD. After the DMD, the laser spectrum is spatially dispersed to different angles. After the collimating lenses, an objective lens recombines the different spectral components separated by the DMD. Temporal focusing is thus achieved at the focal plane of the objective lens as a thin light sheet of $2\sim 5\text{ }\mu\text{m}$ thick and $50 - 400\text{ }\mu\text{m}$ in diameter depending on the optics. The spectral components will diverge again after leaving the objective's focal plane. In other words, ultrashort pulses only retain the narrow 100 fs pulse width at the focal plane of the objective lens and broadened everywhere else, thereby achieving depth discrimination.

Figure 2 presents parallel laser patterning results performed by the TFLMS. In Figure 2A, a “CU” (Chinese University of Hong Kong) pattern was directly micro-machined within a pollen sample by photo-bleaching. In Figure 2B, “CUHK” patterns of different sizes were directly micro-machined on a nickel target; the dark indent in the background represents the processing area ($\sim 50 \times 50\text{ }\mu\text{m}^2$) with laser power of 1 mW at the focal region. In Figure 2C, a set of alphabets were micro-machined on the nickel target with a resolution approaching the diffraction limit, i.e. 500 nm. The grain boundary of the nickel target can be clearly observed in the SEM image.

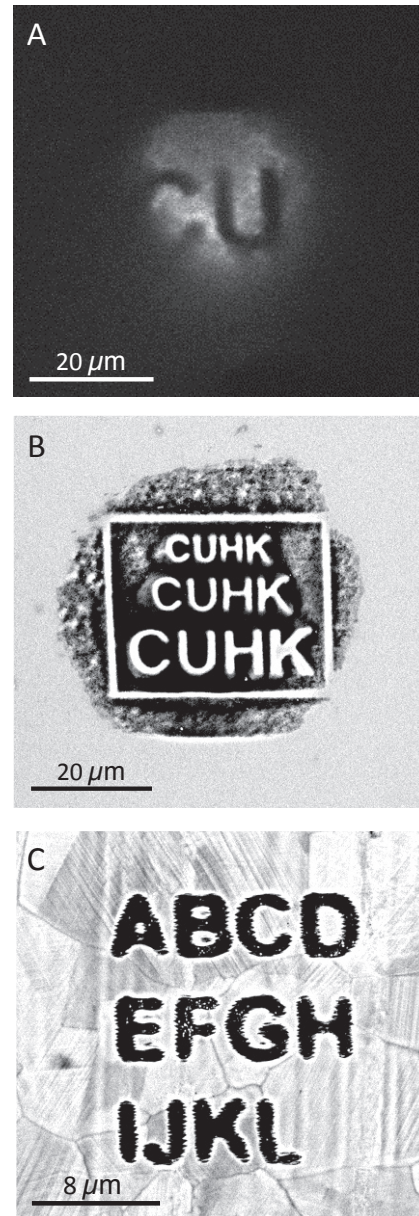


FIGURE 2. A: Optical image of a “CU” pattern written directly within a pollen sample; B & C: Direct area patterning results on a nickel target, demonstrating close to diffraction limit resolution. The dark indent in B shows the processing area ($\sim 50 \times 50\text{ }\mu\text{m}^2$).

Figure 3 presents a pulse-by-pulse direct area patterning sequence on a silicon substrate, where the laser amplifier was pumped with a 16.1 A current (corresponding to 0.19 J/cm^2 at the substrate surface). In the experiment, a total of 9 pulses were used to engrave the different alphabets inside an $80 \times 60\text{ }\mu\text{m}^2$ box. It is worthwhile to note that changing the laser power may slightly affect the final pattern depth as it

increases/decreases the thickness of the light sheet. Since the repetition rate was 1 kHz, the entire fabrication progress was completed within 9 milliseconds (equivalent to $\sim 1 \text{ mm}^2/\text{s}$). Higher repetition rate femtosecond laser will further increase the throughput, though heat accumulation effect may need to be considered. In contrast with a point scanning system, it will require several minutes to pattern the box with the same resolution. Moreover, the point-scanning system can be more expensive due to the need of precision stages and synchronization hardware.

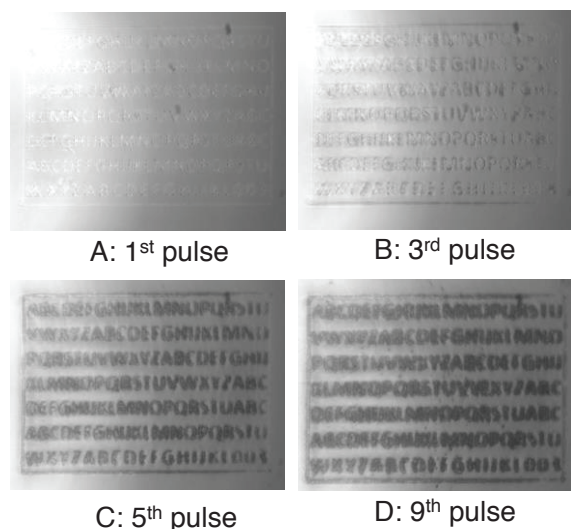


FIGURE 3. Alphabets patterned on a silicon substrate in an $80 \mu\text{m} \times 60 \mu\text{m}$ box at the first (A), third (B), fifth (C), and ninth pulse (D), demonstrating direct area patterning

The next experiment was devised to demonstrate the optical sectioning capability of the TFLMS. The optical sectioning capability benefits the fabrication of opaque materials by generating a flat surface. In this experiment, the TFLMS was used to create a circular pattern of 4 micron deep on a silicon substrate. The depth control capability is attributed to the highly confined laser power distribution, i.e. the thin light sheet. An AFM image and line-scan profile of the result are shown in Figure 4A and 4B, respectively, indicating that the bottom of the circular pattern is flat (vs. the curved bottom profile produced by a Gaussian beam). Interestingly, we also find the roughness of the flat bottom to be relatively smooth (RMS $\sim 60\text{nm}$). A possible explanation of this phenomenon is that the temporally focused light sheet has a flat and uniform laser power threshold boundary. These results are difficult to

achieve by other laser fabrication methods, e.g. laser projection, as the laser power is Gaussian distributed, resulting in concave surfaces. Accordingly, a flat top beam shaping unit is not required in the TFLMS which further simplifies the system.

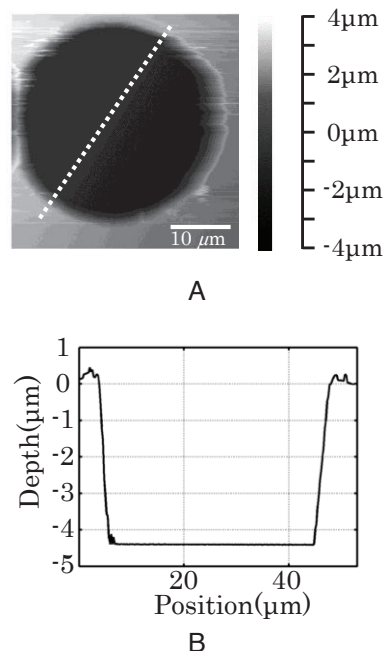


FIGURE 4. A $4 \mu\text{m}$ deep hole patterned on silicon by the TFLMS. A: Top view of the hole scanned by an AFM. The scale bar is $10 \mu\text{m}$; B: Cross-sectional profile of the hole, indicated by the dash line in A, where the RMS surface roughness of the bottom is $\sim 60 \text{ nm}$.

PARALLEL METAL SINTERING/METLING

In this section, we use the TFLMS to study the metal sintering and melting processes and investigate whether temporal focusing technique can be used to improve the resolution and throughput of the current metal additive manufacturing process, i.e. 3-D metal printing. Like laser micro-machining, current 3-D metal printing technology is based on selective laser sintering (SLS) or selective laser melting (SLM), i.e. to scan a laser beam in space to sinter or melt powder materials, which is a sequential process that is inherently slow [2]. In addition, the scanning speed is restricted by the laser power to avoid generating discontinuous scan tracks or forming of large drops, i.e. balling effect [3].

The 3-D resolved temporal focusing approach may have benefits in both resolution and throughput compared with current metal additive

manufacturing systems for the following reasons: (1) patterns are created in parallel over the processing area, e.g. $400 \times 400 \mu\text{m}^2$ within 1 - 10 milliseconds, (2) the axial resolution will be improved as the optical energy is highly confined within the light sheet of 2~5 micron in thickness, and (3) the lateral resolution is limited by diffraction, i.e. 400 nm for the regenerative amplifier.

In the sintering and melting experiments, the laser source is a Ti-Sapphire femtosecond oscillator (Coherent, Chameleon Ultra II) with a power output of 3.5 W at 800 nm, and a repetition rate of 80 MHz. To achieve high patterning resolution, metal (iron) powders of an average grain size of 200 nm were used. (Note that commercial systems use metal powders of an average grain size of 10s microns.) As shown in Figure 1, an epi-illumination light source and a CCD camera are coupled into the light path to monitor the process in real time. Also, the processing time can be precisely controlled by changing the patterns on the DMD, which has a pattern rate of 4.2 kHz. Accordingly, a mechanical shutter is not needed in our system.

Figure 5 presents the temporal focusing sintering results. In this experiment, we used low laser intensity (2.5 W/mm^2) to sinter iron powders in a parallel fashion, producing a $\sim 2 \times 2 \text{ mm}^2$ thin iron layer, shown in Figure 5A. As shown in Figure 5B, the SEM image of the metal layer presents interesting results of controlled porous structures, which may find important applications in optical sensing, e.g., plasmonic sensors, and energy storage, e.g., electrodes of super-capacitors.

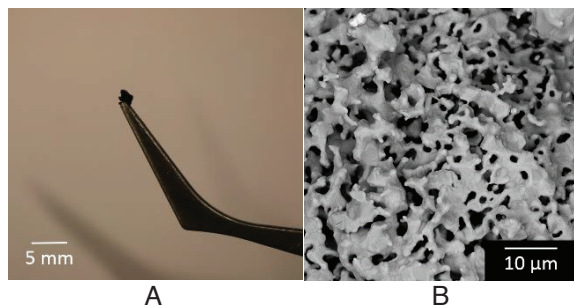


FIGURE 5. A: A sintered iron powder layer; B: SEM image of the iron powder structure

As sintered metal structures may not have the required structural strength for certain 3-D metal printing application, we performed experiments at a higher laser intensity level (17.5 W/mm^2) to

investigate whether temporal focusing can melt a layer of metal powders. Figure 6 presents the results, where a clear junction between the powder and fully melted area can be observed under an SEM. We also find some sintered metal powders nearby the junction due to thermal diffusion.

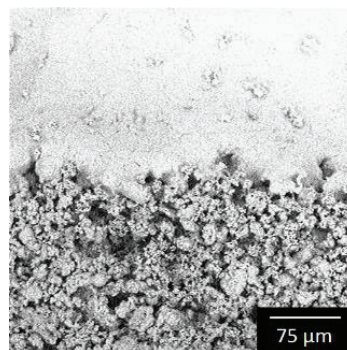


FIGURE 6. SEM image of the melted metal layer

In the next experiment, we programmed two different patterns, i.e. “X” and “□”, to the DMD to process an area of $420 \times 260 \mu\text{m}^2$. The results are presented in Figure 7A and 7B, confirming that the TFLMS can be used for parallel metal sintering and melting with a resolution of approximately 30~50 μm .

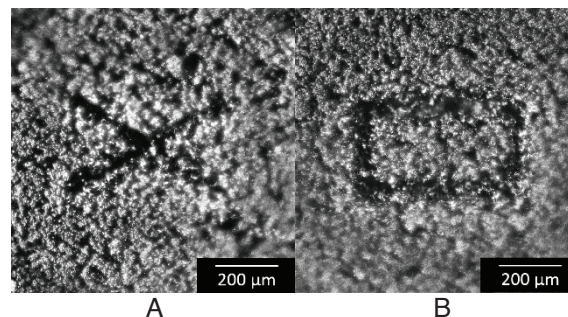


FIGURE 7. Parallel laser sintering/melting results with patterns defined by the DMD. A: “X” pattern; B: “□” pattern

Although the metal sintering/melting process has a lower resolution than the micro-machining process (500 nm), it is still substantially better than the current SLS and SLM processes, which typically have a resolution of 100s microns. Other factors that contribute to the lower than expected resolution include, (1) thermal diffusion, and (2) nanoparticle agglomeration. The resolution may be improved by fine-tuning the processing parameters, including average laser power and repetition rate.

CONCLUSION

In this paper, we present the development of the TFLMS for both laser-machining and metal sintering/melting processes. The TFLMS is a promising solution for high-resolution (500 nm), high-throughput (1 mm²/s) laser micro-machining based on femtosecond lasers. The fast pattern switching capability of the DMD presents great potential to perform fast 3-D engraving on both transparent and opaque materials. The patterning resolution and speed can be further increased by using femtosecond pulses of shorter wavelengths. For metal sintering/melting experiments, the TFLMS demonstrates 30~50 μm patterning resolution over an area of 420 $\times$ 260 μm^2 , presenting significant advantages over conventional point-scanning based metal additive manufacturing processes.

ACKNOWLEDGMENT

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REFERENCES

- [1] Oron D, Tal E, and Silberberg Y. Scanningless Depth-Resolved Microscopy. *Optics Express*. 2005; 13(5): 1468-1476.
- [2] Frazier W E. Metal additive manufacturing: A review[J]. *Journal of Materials Engineering and Performance*, 2014, 23(6): 1917-1928.
- [3] Tolochko NK, Mozzharov SE, et al. Balling Processes During Selective Laser Treatment of Powders. *Rapid Prototyping Journal*. 2004; 10(2): 78-87.

USING CONTROLLED THERMAL EXPANSION FOR TOOL ALIGNMENT IN ULTRA-PRECISION MILLING

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INTRODUCTION

Diamond machining is considered a versatile technique for generating optical surfaces and microstructures [1, 2]. Depending on the shape and complexity of the part, different processes may be applied. For machining of freeform surfaces, diamond milling is a common choice [3, 4]. Despite the limited spectrum of machinable materials for diamond machining in general, the inevitable long processing time is a major disadvantage of this technology. This aspect is closely connected to the precision of the machined parts: due to the tolerances in machining and assembly, using a tool holder with multiple tool-inserts leads to different circular paths of the individual cutting edges. Considering a kinematic roughness of an optical surface to be $R_{kin} < 10 \text{ nm}$, the applied cutting edges may be involved in the material removal, but not in the generation of the surface topography (i.e. $\Delta r_{fly} > R_{kin}$). If the difference is too large ($\Delta r_{fly} > h_{cu,max}$), the inward cutting edge will even not be engaged in material removal at all (see Figure 1).

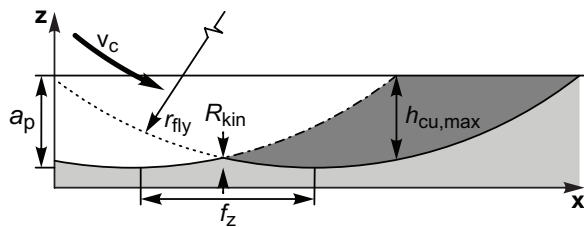


FIGURE 1. Kinematic roughness and minimum undeformed chip thickness in diamond fly-cutting

Thus, circumferential diamond milling generally is conducted as a fly-cutting operation, i.e. using only a single diamond tool, which guarantees that the effective cutting edge describes a defined fly-cut radius r_{fly} . For applying multiple diamond tools in fly-cutting operations, a high-precision alignment of the cutting edges in radial direction is mandatory. While the level of precision required for optical surfaces cannot be achieved with purely mechanical means, electri-

cal actuators (e.g. piezos) depend upon additional components for power supply and information exchange which increase mass and volume of the tool holder.

THERMAL ACTUATOR FOR PRECISION ALIGNMENT OF DIAMOND TOOLS

As a novel approach, the alignment of multiple diamond tools to a common fly-cut-radius is achieved by utilizing controlled thermal expansion of the tool holder. The basic idea is to locally heat the substrate material by electromagnetic radiation in order to induce a thermal expansion shifting all tools to the orbit of the tool with the largest fly-cut-radius (Figure 2).

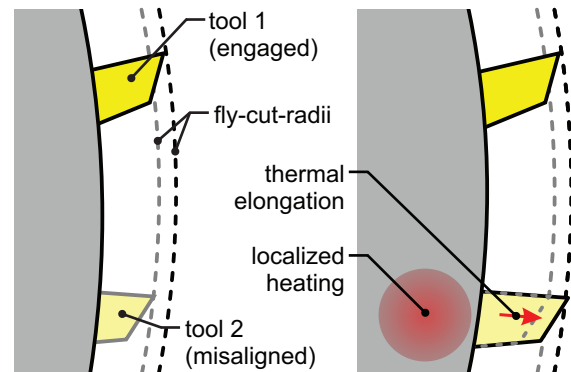


FIGURE 2. Local heating of tool holder to induce radial shift of the cutting edge

Due to the contactless transfer of heat, this solution has several advantages over other electromechanical or purely mechanical actuators (e.g. piezo or screw-driven mechanisms). Most importantly, no additional disturbances (e.g. due to tolerances of the interfaces or clamping forces) are induced to the process, as it would be the case, if components for power supply and control needed to be integrated on the tool holder, i.e. increasing its weight and volume. Furthermore, once a specific amount of heat is introduced to the substrate, it takes a considerable amount of time to dissipate, which means that the position of

the tool can be kept stable with minimal efforts of the control system (i.e. feeding small amounts of additional energy over long periods of time, see Figure 3). Challenges, however, are to limit the elongation of the substrate to the desired (radial) direction by designing an appropriate actuating mechanism and to calculate the required energy intake based on the characteristic elongation of the actuator.

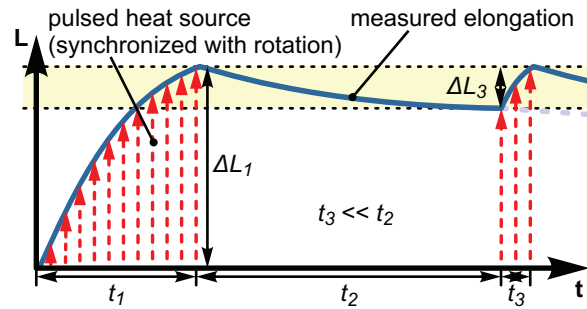


FIGURE 3. Inducing defined elongations on rotating tool holder by triggering the heat input

In this approach, the thermal actuator is designed as a $15 \times 30 \times 7.5 \text{ mm}^3$ steel beam that is connected to the tool holder via struts and flexure hinges (Figure 4). While the details of the actuator design have been subject to a previous publication [5], this paper focuses on the characterization of the actuating mechanism in a static (i.e. non-rotating) test stand, in which the actuating mechanism is irradiated via a high power infrared light source ($P_{\text{LED}} = 0.43 \text{ W}$, $\lambda = 850 \text{ nm}$). Initial experiments have shown that an elongation of approximately 56 nm is achieved after 10 s of illumination [5].

ACTUATOR REQUIREMENTS FOR APPLICATION IN ULTRA-PRECISION MILLING

To be successfully applied in an ultra-precision milling setup, the thermal actuator has to meet specific requirements concerning achievable elongation, precision and temporal stability. Considering the accuracy of the pre-setting mechanism [5], the thermal actuator needs to achieve a maximum elongation of $\Delta r_{\text{fly,max}} \approx 1 \mu\text{m}$. As stated above, the precision has to be smaller than the kinematic roughness of typical optical surfaces, i.e. $\Delta r_{\text{fly,min}} \leq 10 \text{ nm}$.

The requirements concerning the temporal stability are imposed by the dynamics of the cutting process. As the LED light source is fixed at a specific radial and angular position and has a narrow illumination angle, only limited time is available for

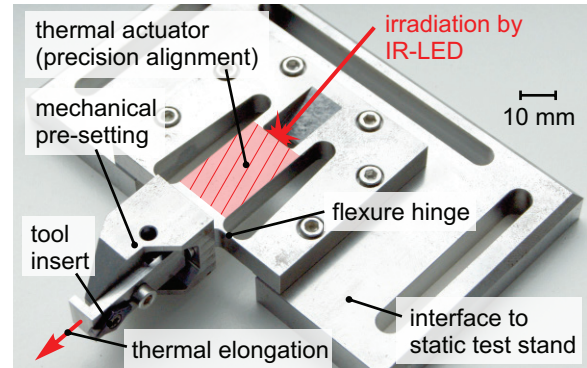


FIGURE 4. Tool holder for thermal alignment of diamond cutting edges on a static test stand

heat transfer to the rotating actuator. The target face at the back of the actuator is designed to rotate at a radius of $r_t = 30 \text{ mm}$ and has a width of $w_t = 10 \text{ mm}$. Thus, at a (typical) spindle speed of $n = 1200 \text{ min}^{-1}$, only $t_i = 2.65 \text{ ms}$ are available for illumination. Furthermore, the target position of the actuator must be maintained for at least one revolution of the tool holder ($t_n = 50 \text{ ms}$), until the target face re-enters the scope of the LED, despite cooling of the actuator, e.g. due to forced convection.

SURFACE PROPERTIES AND HEAT INPUT

Considering the limited time for heat input, the target face needs to be able to absorb as much energy as possible to increase the effectiveness of the actuator. To determine the influence of the surface properties, experiments with varied roughness and dyed target face were conducted. Therefore, steel samples of the same size of the actuator were ground to different roughnesses and illuminated by the IR-LED for $t_i = 100 \text{ s}$ (Figure 5).

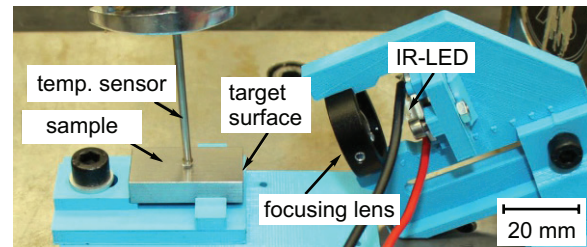


FIGURE 5. Setup for measurement of heat input

By measuring the temperature T_a of the specimen, the gradient of temperature change $\vartheta' = \Delta T_a / t_i$ was calculated for each sample. These experiments were repeated using the same sam-

ples, but with target face colored by black dye, to further increase the heat input.

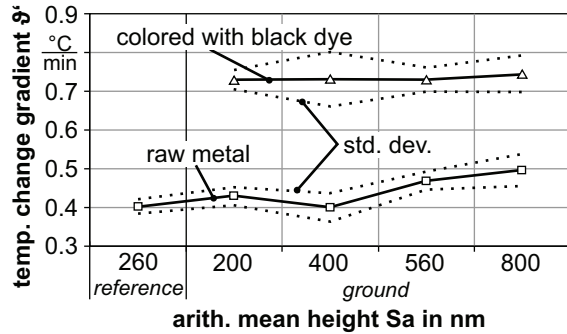


FIGURE 6. Influence of surface roughness and color on heat input

The results shown in Figure 6 indicate that in case of the raw-metal surfaces, the gradient of temperature change increases with increasing roughness from $\vartheta' = 0.4^{\circ}\text{C min}^{-1}$ for the low-roughness and untreated surfaces to $\vartheta' = 0.5^{\circ}\text{C min}^{-1}$ for the high-roughness surface. By coloring the face with black dye the surface, the heat input is significantly increased. For all samples, a similar gradient of $\vartheta' = 0.73^{\circ}\text{C min}^{-1}$ was calculated.

INFLUENCE OF FORCED CONVECTION

Additional experiments were conducted in order to evaluate the effect of forced convection on the resulting thermal elongation of the actuator. Here, the target face of the thermal actuator was illuminated in a static test stand while measuring the temperature at the center of the actuator T_a and the resulting thermal elongation at the tool-interface Δl_a (Figure 7).

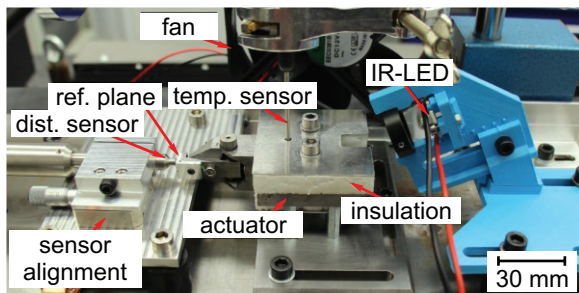


FIGURE 7. Static test setup with active cooling and thermal insulation

For simulating the effect of forced convection due to the rotation of the tool holder, the static test stand was equipped with a $\varnothing 120\text{ mm}$ -fan. The fan delivers a throughput of $234\text{ m}^3\text{ s}^{-1}$, resulting in a

flow speed of 5.7 m s^{-1} , equal to the mean rotational speed of the tool holder at $n = 1200\text{ min}^{-1}$. In order to allow for an even circulation of the actuator, it was mounted on 20 mm high spacers.

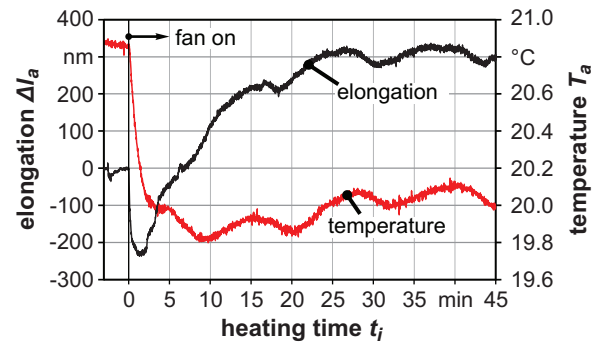


FIGURE 8. Influence of forced convection on thermal elongation

Figure 8 shows an exemplary measurement of elongation and temperature. It can be seen that after switching on the fan, the forced convection results in an abrupt drop in temperature from about 20.9°C to $T_a \approx 20^{\circ}\text{C}$. At the same time, the front face is retracted by $\Delta l_a \approx -200\text{ nm}$. In the following minutes, the elongation of the actuator increases again to $\Delta l_a \approx 300\text{ nm}$. This, however, is not accompanied by an equal rise in actuator temperature. The reason for this is yet unknown. As no drift in expansion nor in temperature measurement was observed, an explanation for this might be the overall cooling of the setup induced by the fan, causing all parts to shrink and thereby influencing the measured elongations. This theory is supported by the fact that no hysteresis is observed after switching off the fan again, i.e. Δl_a returns to 0 nm. As a consequence, all measurements including forced convection required an acclimatisation phase of several minutes to reach a steady state of actuator and metrology frame.

COMPARISON OF DIFFERENT SETUPS

For evaluating the achievable elongation and actuating speed, three different setups for the LED and actuator were analyzed. The first setup comprises the actuator itself and the LED directly irradiating the target face at the back of the actuator. In a second setup, a lens was introduced in order to focus the IR-radiation of the LED to the target face of the actuator. The third setup added an insulation covering top and bottom of the actuator, in order to minimize heat-loss due to convection. The results of elongation measurements for the three setups are shown in Figure 9.

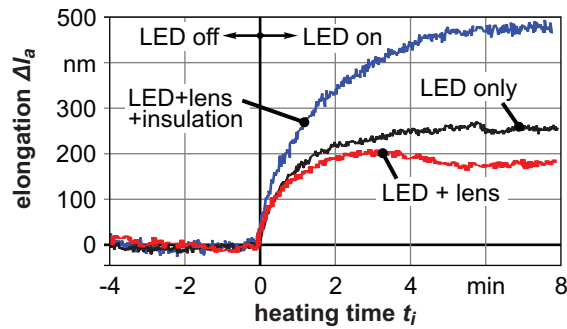


FIGURE 9. Static test setup with active cooling, w and w/o lens and thermal insulation

The experiments have shown that it is possible to reach an elongation of >200 nm with all setups, despite the influence of forced convection. For the first two setups, the elongation reaches a steady state after approx. $T_a = 2$ min of continuous irradiation. It was found that the addition of a focusing lens does not increase the heat intake to the actuator. The resulting elongation is even lower than in the lens-free setup. The reason for this is supposedly caused by the lens itself, focusing the radiation on the one hand, but also absorbing a portion of the radiant flux. Introducing a thermal insulation to the actuator results in an almost doubled elongation ($\Delta l_a \approx 500$ nm) compared to the other setups. In this case, a steady state is reached after about $T_a = 5$ min.

DESIGN OF A SIMPLE CONTROL LOOP

On the basis of the static test stand, a simple feedback control loop for setting the position of the thermal actuator was derived. It acts as a two-point control, switching the LED on and off, if the measured elongation falls below a defined limit or exceeds a specific threshold. Initial results for position control are depicted in Figure 10. From the initial elongation -20 nm two step changes to 40 nm and 0 nm are executed, showing that a high accuracy and position stability can be achieved.

SUMMARY AND OUTLOOK

For the application of multiple diamond tools in ultra-precision milling, a high-precision actuating mechanism for aligning the orbit of the cutting edges is mandatory. It has been shown that a thermal actuator is able to fulfill the tight requirements for optics manufacturing. In this paper, the actuator is characterized according to the achievable heat input and its reaction to forced convection in a static test stand. These results deliver

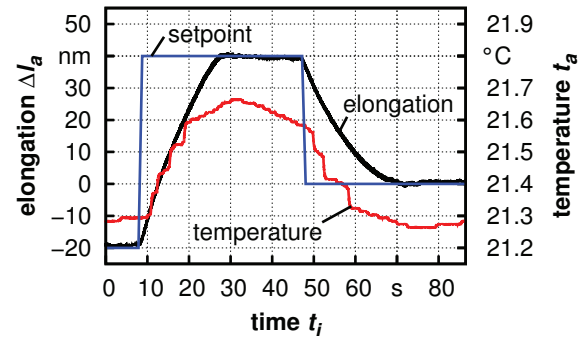


FIGURE 10. Commanded elongation using position feedback loop and two-point controller

the basis for developing a feedback control loop for setting defined elongations.

The next steps will be to optimize the control loop and transfer the actuator to a spindle setup and later to an actual diamond milling process for evaluation under real cutting conditions.

ACKNOWLEDGMENTS

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REFERENCES

- [1] Dornfeld D, Min S, Takeuchi Y. Recent Advances in Mechanical Micromachining. CIRP Ann Manuf Techn. 2006;55(2):745–768.
- [2] Shore P, Cunningham C, DeBra D, Evans C, Hough J, Gilmozzi R, et al. Precision engineering for astronomy and gravity science. CIRP Ann Manuf Techn. 2010;59(2):694 – 716.
- [3] Brinksmeier E, Preuß W. Micro-machining. Phil Trans R Soc A. 2012;270:3973–3992.
- [4] Fang FZ, Zhang XD, Weckenmann A, Zhang G, Evans CJ. Manufacturing and measurement of freeform optics. CIRP Ann Manuf Techn. 2013;62(2):823–846.
- [5] Schöнемann L, Mejia P, Riemer O, Brinksmeier E. Design and evaluation of a thermal actuator for ultra-precision machining. In: Proc. LAMDA MAP 11th Int. Conf.; 2015. p. 320–330.

SYSTEM BEHAVIOUR OF A MULTIPLE OVERCONSTRAINED COMPLIANT FOUR-BAR MECHANISM

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ABSTRACT

A method is proposed to identify the critical misalignment of a multiple overconstrained mechanism. This misalignment indicates the limit of desirable behaviour. The proposed method is compared with a multibody simulation of a compliant four-bar mechanism with three overconstraints. Subsequently, both analyses are compared to an experimental setup.

The proposed method compares well to the multibody model. The proposed method and multibody simulation match qualitatively with measurements of the four-bar mechanism. However, the measured critical misalignments are about 30% larger in the experiment. As such, the proposed method is able to compute the critical misalignment of the four-bar mechanism.

INTRODUCTION

Compliant mechanisms are used in precision mechanics to mitigate the effects of play, among others. In theory, their behaviour is predictable and deterministic. To obtain deterministic behaviour in an assembled mechanism, the principle of exact constraint design is often used.

Misalignments increase the internal stress in the mechanism. In an exactly constrained design the increase of internal stress due to misalignment is minimised. Yet these mechanisms can also suffer from limited load carrying capability and asymmetric and complex design. Overconstrained designs do not suffer from these drawbacks, but have a relatively small tolerance on misalignment. In a flexure-based mechanism this will cause unwanted static and dynamic system behaviour, such as buckling and change in support stiffness. The deterministic behaviour is therefore no longer guaranteed [1, 2].

The consequences of misalignment on a once-overconstrained parallel leaf spring guidance were investigated by Meijaard et al. [1]; a small misalignment in the overconstrained direc-

tion caused buckling and significant changes in the support stiffness and dynamic system behaviour.

This article aims to find a computationally inexpensive method to ascertain the limits of the deterministic behaviour of an overconstraint compliant mechanism by determining the critical misalignment. As such, the work outlined in this article can be used as a guideline to estimate the manufacturing and assembly tolerances of an overconstrained flexure-based mechanism.

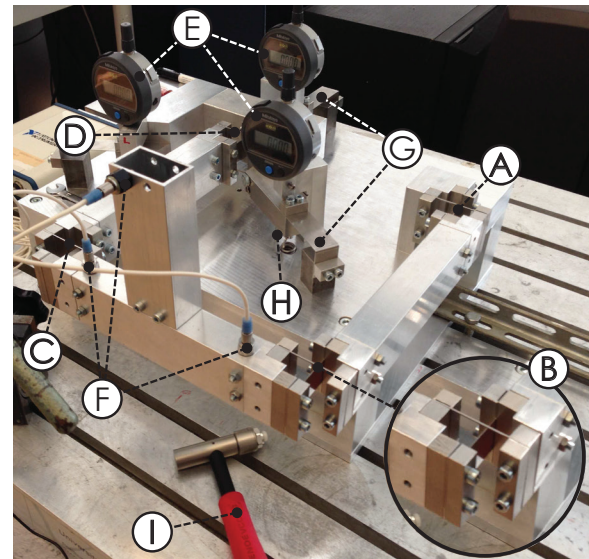


FIGURE 1. Photo of the experimental four-bar mechanism, with the cross pivot flexures (A,B,C,D) between the bars, dial gauges (E) on the manipulator and accelerometers (F) on the effector, 3 in-plane folded leaf spring constraint elements (G), 3 out-of-plane adjustment screws for the misalignment (H) and a modal hammer (I) to excite the mechanism.

SYSTEM DESCRIPTION

The four-bar mechanism under consideration is shown in figure 1. The four hinges are equal, both in dimension and orientation. A hinge is de-

signed to be exactly constrained; it has no over-constraints and one compliant rotation. A hinge consists of one leaf spring and two slender rods (inset B of figure 1); this cross pivot flexure is compliant for rotation along the vertical axis at the intersection of the leaf spring and the slender rods.

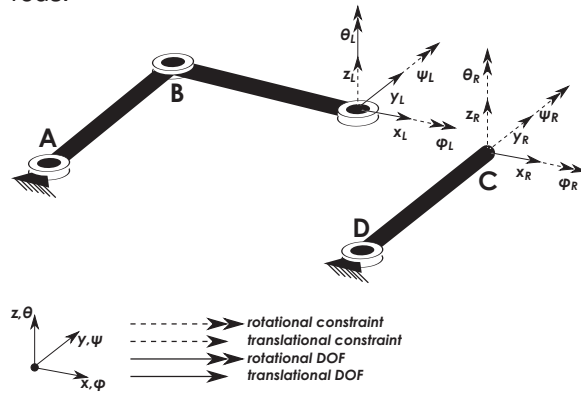


FIGURE 2. The kinematics of a four-bar mechanism illustrated by opening the loop at hinge C; the constraints of the left and right side are shown.

These flexures are used as hinges in conjunction with three rigid bars and the base to create the four-bar mechanism. The resulting mechanism has one degree of freedom (DOF) and three over-constraints. These can be illustrated by opening the mechanism at hinge C (figure 2). The in-plane directions of the mechanism are x , y and θ , whereas the out-of-plane directions are z , ϕ and ψ . Each hinge constrains five directions and leaves one free. The left side of the loop has three hinges and therefore has all in-plane DOFs. The right side of the loop has one in-plane DOF, the combined x_R and θ_R motion, since it only contains one hinge. This is the DOF of the mechanism. The three constraints from the left, z_L , ϕ_L and ψ_L , are equal to the constraints z_R , ϕ_R and ψ_R from the right side. Therefore the mechanism is three-times overconstrained [2].

The effects of these overconstraints can be shown by manipulating the three corresponding misalignments, the z -, ϕ - and ψ -misalignments, at one end of the mechanism; in this case it is the centre of compliance of hinge D. This manipulated point allows for independent control of each misalignment. The three independent misalignments can be controlled by three adjustment screws (H in figure 1). Gauges are used (E in

figure 1) to measure the displacements. The specific misalignment at the manipulator (between D and G in figure 1) can be inferred from these displacements.

METHOD

A method is presented to provide insight into the possible effects of overconstraints as well as to simplify the problem of determining the buckling behaviour. The loop buckling analysis presented in this section divides the analysis of the buckling behaviour into two problems. First the forces in the mechanism due to a misalignment must be determined; second the buckling loads of the flexible elements of the mechanism are determined. The combination of these two parts leads to a relation between misalignment and the buckling behaviour of the flexures.

The relation between misalignment and stresses and forces in the mechanism is analysed. To that end, the equivalent compliance of the mechanism at the manipulator is determined. Several assumptions are made to calculate this equivalent compliance for this case. The four-bar mechanism is in its neutral position and will not deflect significantly if misalignment is added. Furthermore, the bars are considered infinitely stiff and the hinges are considered infinitesimally small. This allows for the computation of the equivalent stiffness of the mechanism as a serial loop of rigid bars and compliant hinges, which can deform in six directions.

Since the bars are rigid, the hinges are the only components in the mechanism that will be able to buckle. By means of free body diagrams the forces acting on each hinge are then determined from the forces acting on the mechanism at the manipulator. These resultant force vectors at the hinges are used in an analysis of the buckling behaviour of the hinges.

Due to the topology of the hinges, the compliance of the hinges can be lumped to the centre of compliance of the flexures. In this case this is at the intersection of the leaf spring and the slender rod flexure. The centre of compliance is the point in certain flexures where a force or moment in a certain direction will only cause a displacement in that same direction [3].

A finite element approach is used to determine the equivalent compliance. The hinges are treated as an infinitesimally small spring element

which can deform in all six directions. The bars of the mechanism are rigid. A set of nodal coordinates $x^{(k)}$ describe the locations and orientations of the nodes of an element k . Deformation coordinates $\epsilon^{(k)}$ are used to describe the deformation of an element. These are invariant for the rigid bars; for the hinges they describe the elastic deformation due to forces [2].

The compliance matrix is the same for every hinge. The force vector $f^{(h)}$ is specific for a hinge. Expressions for all nodal coordinates are determined using the finite element approach. These expressions are functions of the forces in the mechanism. The same holds for the coordinates of the end node q of hinge D; this node is the manipulated point. The forces at the hinges are related to an input force vector at that node; the forces at hinges, $f^{(A)}$, $f^{(B)}$ and $f^{(C)}$, are written as a function of the forces at hinge D, $f^{(D)}$. Consequently, the resulting coordinate vector $x_q^{(D)}$ is a function of the input force vector $f^{(D)}$. The equivalent compliance of the mechanism at the coordinate $x_q^{(D)}$ is the Jacobian matrix of that vector function. The change in the nodal coordinate vector $x_q^{(D)}$ describes the added misalignment.

The three overconstraints in the mechanism lead to forces and moments on the hinges which can cause them to buckle. The buckling multiplier of a hinge is the ratio between the critical buckling load in a direction and the magnitude of the applied force vector in that same direction.

The equivalent stiffness at the manipulator is used to determine the explicit forces at the manipulator; these are related to specific amounts of misalignment. For each applied misalignment the resultant forces at the hinges are calculated. The three possible misalignments yield three explicit force vectors at the manipulator. There are four hinges in the mechanism; in total there will be twelve buckling cases.

Initially the resultant forces are used in a multi-body simulation of a single hinge. The program SPACAR [4] can then determine the buckling multipliers and modes of the specific buckling case. The twelve unique buckling cases are all simulated.

The buckling loads were also estimated using the classical buckling equations [5]. The resultant force vector can be factored into two components:

the part responsible of the lateral buckling of the leaf spring and the part responsible of the axial buckling of one of the slender rods. The corresponding component of the resultant force vectors is compared to the estimated critical buckling loads in the same direction to determine the buckling multipliers.

A full elastic multibody simulation of the mechanism serves as a reference and is compared to the loop buckling analysis. Elastic beams are only used for the flexures of the hinges. Each hinge is modelled with twelve flexible beams; four for the leaf spring and four for each of the two slender rods. The torsional stiffness of the leaf spring is stiffened at the end to take into account constraint warping effects [6].

METHOD RESULTS

The results of the two methods are shown in table 1. The lowest critical misalignment of a hinge is listed in bold: at that specific misalignment at the manipulator that hinge will buckle. The difference between the methods is probably caused by an incorrect estimate of the effective lengths of the flexures. The equivalent length is hard to estimate; the buckling problem of the flexures has elastic constraints.

TABLE 1. The critical misalignments obtained with the simulation (sim) and the classical equations (eq) with respect to the hinges. The lowest misalignments are listed in bold.

hinge		A	B	C	D
$z(mm)$	eq	2.15	2.15	2.15	2.15
	sim	2.33	2.33	2.33	2.33
$\phi(mrad)$	eq	11.5	21.5	21.5	11.5
	sim	11.9	23.9	23.9	11.9
$\psi(mrad)$	eq	15.1	15.1	6.28	6.28
	sim	16.7	16.7	6.99	6.99

The mechanism simulation shows agreement with the hinge simulation with respect to the ϕ - and ψ -misalignments (table 2). The buckling mode is consistent with both the hinge simulation and the classical buckling equations. For instance, it can be seen in table 1 that hinges C and D buckle first due to a ψ -misalignment; the mechanism simulation also shows this behaviour.

TABLE 2. The calculated and simulated buckling misalignments of the mechanism

misalignment	$z(mm)$	$\phi(mrad)$	$\psi(mrad)$
mechanism sim	2.01	12.0	7.22
hinge sim	2.33	11.9	6.99
classical eq.	2.15	11.5	6.28

EXPERIMENT

The dynamic behaviour is measured with the help of a modal analysis. The vibrations of the effector are measured with three acceleration sensors; the mechanism is excited with a modal hammer. Frequency response functions are then determined using these four signals. Four vibration modes of the effector are measured. Of these four modes the third mode showed the most illustrative changes in the system behaviour. The third mode is the motion of the effector in the ψ -direction (figure 2). The misalignment was increased until visible buckling occurred. The results concerning the ψ -misalignment are shown in figure 3.

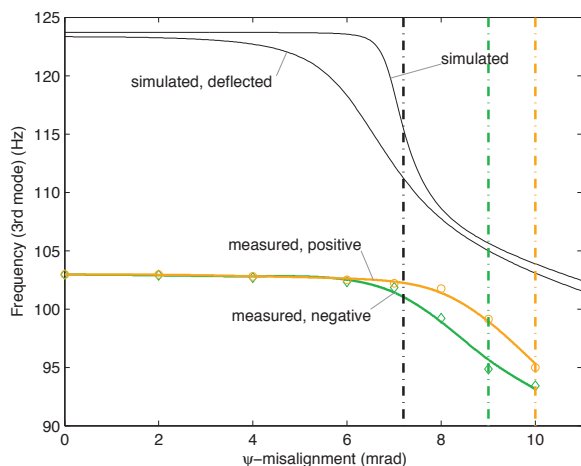


FIGURE 3. Modal profiles of the third mode due to a ψ -misalignment, with the simulated values in black (both negative and positive), the negative measured values in green and the positive measured values in orange. The critical misalignments are the dashed lines, again black for the value found in the simulation, green for the value at negative visible buckling and orange for the positive value.

The mechanism in the simulation was deflected by half the thickness of the leaf spring flexures ($250 \mu m$) to simulate imperfections in the neutral position. A small deflection causes a significant

change in the slope of the frequency change (figure 3). Qualitatively, the dynamic behaviour of the experiment matches with the simulation of the deflected mechanism. The buckling misalignment is higher, and therefore more tolerant, in the experimental setup. The average critical misalignment found in the experiment is about $2.5 mrad$ larger than the simulated and calculated values: about $9.5 mrad$. The hardware imperfections may influence the boundary conditions of the buckling problem of the slender rods. Still, the measured critical misalignment is only 30% higher than the calculated value, which is rather accurate considering the possible hardware imperfections. Also, this specific mechanism displayed only about a 10% decrease in stiffness in the ψ -direction at the critical misalignment.

The modal measurement on the experimental setup gave distinguishable results. The combination of spring steel flexures and steel clamping blocks resulted in a mechanism with a high Q factor.

CONCLUSION

The system behaviour of a multiple overconstrained compliant four-bar mechanism has been investigated. All approaches yielded values for the critical misalignment in the mechanism in the three overconstrained directions. Agreement was found between these approaches. The buckling misalignments could also be adequately estimated with classical methods. The critical misalignments found in the experiment were consistently higher than those found in the theoretical and numerical calculations: they were about 30% higher.

Importantly, the critical misalignments can be obtained without using complex simulations. The values obtained from the loop buckling analysis compare well with multibody simulations. These misalignments can be used to determine the manufacturing tolerances of a mechanism. The approach outlined in this article can be adapted to fit other types of compliant mechanisms.

REFERENCES

- [1] J P Meijaard, D M Brouwer, J B Jonker. Analytical and experimental investigation of a parallel leaf spring guidance. Multibody system dynamics. 2010;23(1):77–97.
- [2] D M Brouwer, S E Boer, J P Meijaard, R G K M Aarts. Optimization of release loca-

tions for small self-stress large stiffness flexure mechanisms. *Mechanism and machine theory*. 2013;64:230–250.

- [3] H J M R Soemers. *Design Principles for Precision Mechanisms*. Enschede: T-Pointprint; 2010.
- [4] J B Jonker, J P Meijaard. SPACAR — Computer Program for Dynamic Analysis of Flexible Spatial Mechanisms and Manipulators. In: *Multibody Systems Handbook*. Springer Berlin Heidelberg; 1990. p. 123–143.
- [5] S P Timoshenko, J M Gere. *Theory of Elastic Stability*. 2nd ed. New York: McGraw-Hill; 1961.
- [6] D H Wiersma, S E Boer, R G K M Aarts, D M Brouwer. Large stroke performance optimization of spatial flexure hinges. In: *Proceedings of the IDECT ASME 2012*. ASME; 2012. p. 1–10.

STUDY OF SOFT TISSUE CUTTING BASED ON A PRECISION VIBRATING BLADE MICROTOME

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In this paper, we present a flexure-based vibrating blade microtome for sectioning soft tissues with submicron level flatness and precision. Cutting mechanism of soft tissues was modeled based on contact and fracture mechanics in order to establish quantitative relationships between the sectioning quality, i.e. flatness, and microtome operation parameters, including vibration frequency, vibration amplitude, and sample feed rate.

FLEXURE-BASED MICROTOME

Previously, we have reported a flexure-based oscillating blade microtome [1] which was integrated with a high-speed multifocal multiphoton microscope for 3-D whole brain imaging at subcellular resolution, commercialized by TissueVision Inc. in 2012. This microtome generates 1 μm out-of-plane vibration when operated at its maximum oscillation amplitude.

In this work, we have developed an ultra-precision microtome to pair with a super-resolution fluorescence microscope with an aim to obtain a connectome, i.e. a detailed wiring diagram of the entire brain, showing all the cells and their synaptic connections. Generating a connectome will require a resolution of 50 nm or better. To achieve such resolution and to map synapses across sliced sections, it requires a microtome to yield a flatness of 100s nanometer root mean square (RMS) and a predictable synapse deformation at the cutting surface. In addition, since deep imaging can result in image degradation, thin sections of 20 microns (or less) are desirable. However, state-of-the-art vibrating blade microtomes for soft tissue cutting can only produce sections of 40 - 100 microns in thickness with an uneven (wave-like) surface of 1-5 microns RMS due to the unwanted out-of-plane blade vibration. This not only makes it impossible to produce ultra-thin tissue sections, but also results in varying signal strength, i.e. brightness, across the imaging field.

Figure 1 shows the CAD model of our new flexure-based microtome, where multiple parallel flexure blades are used simultaneously to constrain unwanted out-of-plane motions to within 200 nm. (For ideal cutting operations, the blade only oscillates in the horizontal plane, with no parasitic motions in the out-of-plane axes.) The traverse blade motion is generated by an electric motor coupled with an eccentric cam. A mechanical filter,

i.e. a flexure, is then used to isolate the non-transverse cam motion from the blade. Figure 2 shows a zoom-in view of the blade assembly guided by two sets of parallel flexures, i.e. Flexure 1 and 2. Flexure 1 consists of symmetric blades that ensure the blade assembly moves linearly along the transverse direction. Flexure 2 is implemented vertically to further increase the out-of-plane stiffness, thereby reducing the out-of-plane parasitic motions. It is worth to note that the microtome is designed in a way such that the center of mass and center of stiffness of the blade assembly are coincident, where the force from the motor is exerted upon. This design minimizes the unwanted system vibration and out-of-plane parasitic motion during the oscillatory cutting process.

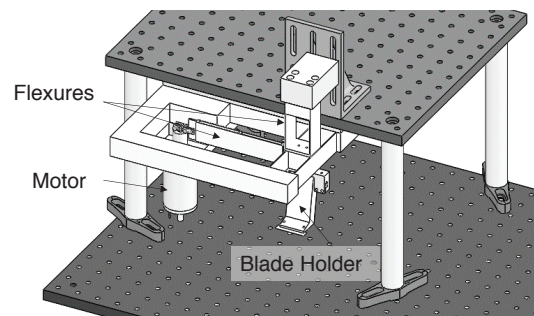


FIGURE 1. 3D model of the assembled microtome

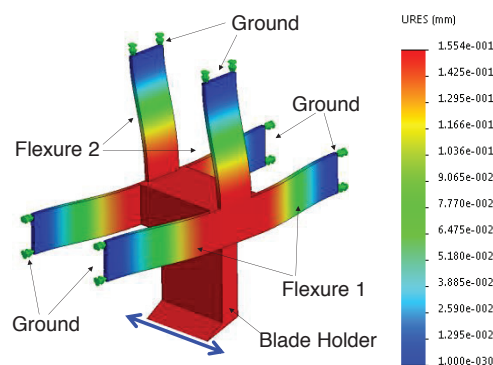


FIGURE 2. Blade assembly with guiding flexures

CUTTING MECHANISM OF SOFT TISSUES

The cutting of soft tissue is a complex process, and it has been known that operating parameters, including blade oscillation amplitude, frequency, blade angle, and sample feed rate, together

determine the sectioning quality and thickness limit. To date, how each parameter can affect the cutting result remains unknown, and the operating parameters for various tissue samples are determined experimentally. In this work, we study this problem with fracture mechanics and the precision microtome.

Soft tissue cutting was studied by Atkins [2], his model explained the effect of the cutting speed ratio (vertical cutting speed to transverse cutting speed) on the cutting force. However, this simplified model did not consider viscosity induced energy dissipation, and it assumed that the transverse cutting speed is constant. As a result, it cannot explain how other microtome operation parameters, e.g. amplitude, frequency and feed rate, affect the sectioning quality.

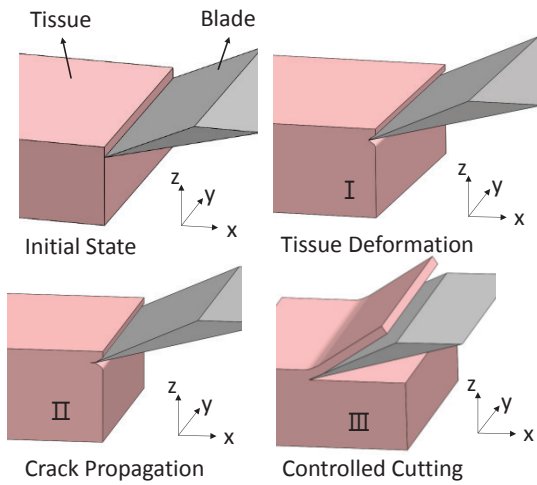


FIGURE 3. Three distinct stages in tissue cutting processes: I: Deformation; II: Crack Propagation; and III: Controlled Cutting

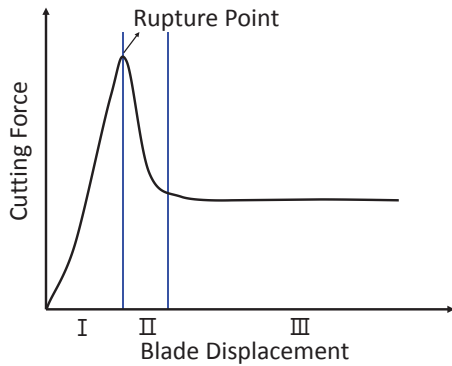


FIGURE 4. Typical cutting force vs. displacement plot during a tissue cutting process

In this work, for the first time we model the oscillatory soft tissue cutting procedure. The model is conceived based on the crack theory. The cutting process is divided into three stages: (1) tissue deformation, (2) tissue rupture, and (3) controlled cutting, illustrated in Figure 3 and Figure 4 [5]. As

shown in Figure 5, using the “ J integral method” in fracture mechanics [3], the crack will extend when the J integral, i.e. the effective energy release rate, around the crack surface reaches the critical value J_c , where J_c is a material parameter. The J integral around a crack tip is defined in Equation 1:

$$J = \int_{\Gamma} (W dy - \vec{T} \cdot \frac{\partial \vec{u}}{\partial x} ds) \quad (1)$$

where J = Effective energy release rate
 W = Strain energy density
 ds = Differential element along the contour
 $\vec{u}$ = Displacement vector at ds
 $\vec{T}$ = Tension vector (traction forces) on the body bounded by Γ
 Γ = Arbitrary counterclockwise contour

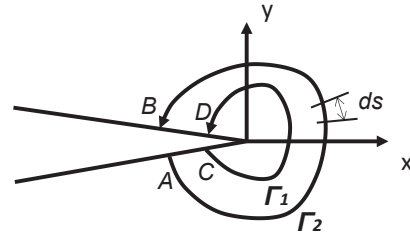


FIGURE 5. J integral contours around crack surfaces

During the tissue deformation stage, the cutting force increases with the blade displacement, and small local cracks begin to develop. By assuming all the work generated by the blade vibration and sample feeding is converted to the strain energy of the tissue, we can obtain Equation 2:

$$\iiint_V W = \int F_x dx + \int F_y dy \quad (2)$$

During the tissue rupture stage, the cutting force suddenly decreases from its peak value. In other words, when the effective energy release rate reaches the critical value, the crack tip starts propagating. At the moment that the rupture occurs, the input work of the blade equals to zero, so that

$$J = \int_{\Gamma} W dy = J_c \quad (3)$$

During the controlled cutting stage, the cutting force is constant when the tissue sample is homogeneous. This is because the strain energy in the tissue remains constant, and the work done by the blade equals to the increase of surface energy induced by the crack propagation. The J integral equals to J_c as the case at the rupture point. Cutting force decreases at this stage because the stress concentration factor, governed by geometric shapes, increases substantially.

Based on experimental observations and past literature, small cutting forces produce tissue sections of better quality. In addition, large cutting forces result in high tissue deformation before the

rupture which is undesired on a precision standpoint. Next, we assume that the stress concentration factor at the rupture point is constant for cases of different feed rates and vibration frequencies. Therefore, in our model, it is assumed cut quality is determined by the maximum cutting force at the rupture point. When the contour integral of strain energy function reaches a critical value J_c , we have:

$$\iiint_V W = W_c \quad (4)$$

$$\int F_x dx + \int F_y dy = W_c \quad (5)$$

Next, we model the reaction force of the soft tissue based on the Voigt model, shown in Fig. 6, for viscoelastic materials. To simplify our model, $k(\omega)$ and $c(\omega)$, i.e. the stiffness and damping coefficient of the material respectively, are assumed to be constants between 20 - 80 Hz, i.e. the microtome operation range. The reaction forces in x and y direction are thus given by

$$F_x = k_1 x + c_1 \dot{x} \quad (6)$$

$$F_y = k_2 y + c_2 \dot{y} \quad (7)$$

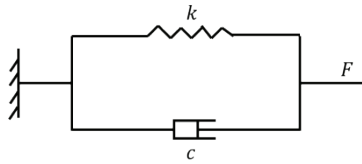


FIGURE 6. The Voigt model for viscoelastic materials is used to model soft tissues.

Accordingly, the maximum cutting force, i.e. the rupture force (F_R), can be calculated as

$$F_R = k_1 x_c + c_1 u \quad (8)$$

where x_c is the tissue displacement from the contact point to the rupture point; $u = dx/dt$ is the sample feed rate. The blade motion can be described as

$$x = ut, y = A \sin \omega t \quad (9)$$

where A is the vibration amplitude; ω is the vibration frequency. Accordingly, the work generated by the blade is

$$W_1 = \int_0^{x_c} F_x dx = \frac{1}{2} k_1 x_c^2 + c_1 u x_c \quad (10)$$

$$W_2 = \frac{x_c}{uT} \left(\int_0^T k_2 A \sin \omega t dA \sin \omega t + \int_0^T c_2 \omega A \cos \omega t dA \sin \omega t \right) = \frac{x_c c_2 A^2 \omega^2}{2u} \quad (11)$$

Substituting Equation (10) & (11) into (5), we have

$$\frac{1}{2} k_1 x_c^2 + c_1 u x_c + \frac{x_c c_2 A^2 \omega^2}{2u} = W_c \quad (12)$$

Replacing x_c with F_R by Equation (8), we have

$$F_R^2 u + F_R c_2 A^2 \omega^2 - c_1^2 u^3 - c_1 c_2 A^2 \omega^2 u - 2W_c k_1 u = 0 \quad (13)$$

Lastly, the solution of Equation (13) is obtained as

$$F_R = \sqrt{c_1^2 u^2 + c_1 c_2 A^2 \omega^2 + 2W_c k_1 + \frac{c_2^2 A^4 \omega^4}{4u^2}} - \frac{c_2 A^2 \omega^2}{2u} \quad (14)$$

From Equation (14), one may quantitatively relate the maximum cutting force, i.e. sectioning quality, to microtome operation parameters, including vibration frequency (ω), vibration amplitude (A), and sample feed rate (u).

EXPERIMENTAL RESULTS

To validate the model, tissue sectioning experiments were conducted based on the new microtome, shown in Fig. 1. To obtain the cutting force, four sets of strain gauges were installed on the blade holder, shown in Fig. 7, to measure the bending strain. Five capacitance probes (Lion Precision, C8/CPL290) were used to monitor the motion characteristics of the blade assembly in a custom-designed metrology frame [1].

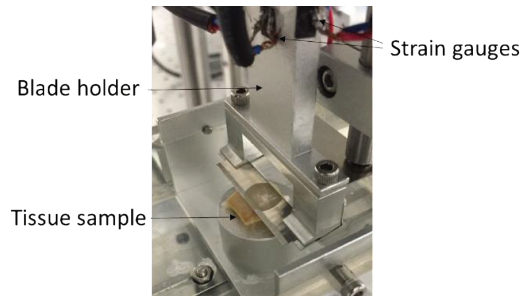


FIGURE 7. Blade holder assembly with four sets of strain gauges installed. The tissue sample is affixed in the water tank on a precision XYZ stage.

Figure 8 presents the out-of-plane parasitic motion at different blade oscillation frequencies, including 20, 40, 60, and 80 Hz, without loading. The results indicate that the parasitic motion has been effectively reduced to 100s nm range by the vertical flexures, satisfying our design goal.

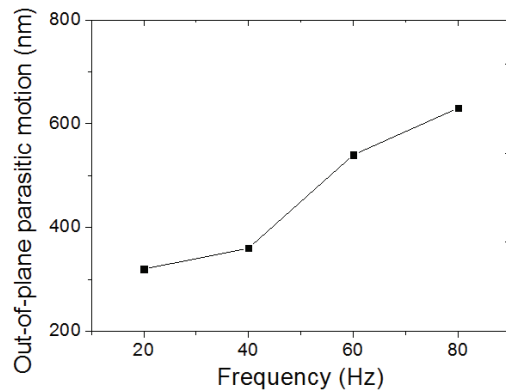


FIGURE 8. Maximum out-of-plane parasitic motion at cutting frequencies of 20, 40, 60 and 80 Hz

Next, we performed sectioning experiments on fixed rat liver tissue, shown in Fig. 9, of different feed rates, vibration frequencies, and thicknesses. Before sectioning, the tissue surface was cut to 10 x10 mm². The vibration amplitude and blade angle were set to 0.15 mm and 35° respectively. The experimental results indicate our microtome can repeatably generate uniform tissue sections of various thicknesses, with a minimal thickness of 10 μm, achieving our design goal.

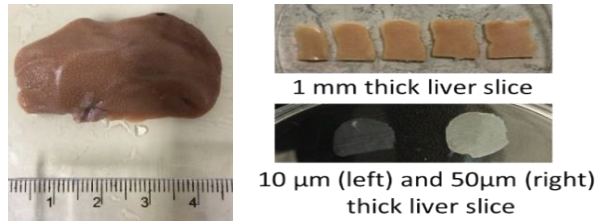


FIGURE 9. Rat liver sample and tissue sections

Figure 10 presents a typical cutting force-displacement plot of liver tissues, where the deformation stage, rupture stage, and controlled cutting stage can be clearly distinguished. We also find small force peaks after the rupture point, which is due to the nonhomogeneity of the tissue sample, i.e. local rupture events still occur in the controlled cutting stage. As the rupture force (F_R) determines the sectioning quality, we plot the rupture force versus different feed rates and vibration frequencies along with our analytic model (Eq. 14) in Fig. 11, which matched well with the measured data. (The material parameters used in the model include: $c_1 = c_2 = 0.0015$ N/mm·s, $W_c = 10$ N·mm, $k_1 = 1$ N/mm.) From Fig. 11, it can be observed that the rupture force decreases with increasing vibration frequency; also, at a given frequency, the rupture force increases with increasing sample feed rate.

Lastly, we characterized the surface profiles of two selected tissue sections, cut with different parameters, i.e. frequency = 20 Hz; feed rate = 0.6 mm/s and frequency = 80 Hz; feed rate = 0.1. Before the surface measurement, tissue samples were frozen to prevent deformation induced errors under the profilometer. The results, shown in Figure 12A and 12B respectively, are well expected but counter-intuitive as minimal out-of-plane parasitic errors of the blade at 20 Hz correspond to a surface roughness of 3.3 μm, and maximal out-of-plane parasitic errors at 80 Hz correspond to a surface roughness of 400 nm. These results have confirmed our assumption that the cutting force determines the cut quality, i.e. minimization of the rupture force ensures high surface quality. Note that for all cutting force measurements, the microtome was programmed to slice tissues into 1 mm sections for consistency.

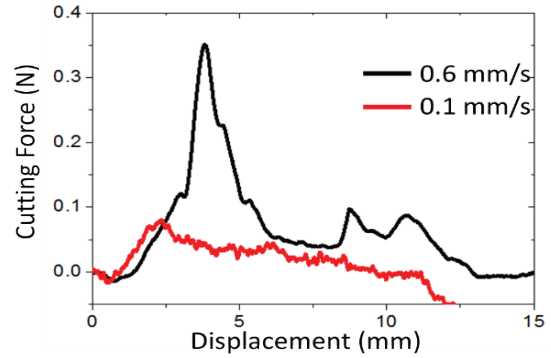


FIGURE 10. Typical force-displacement data of tissue sectioning; vibration frequency = 80 Hz

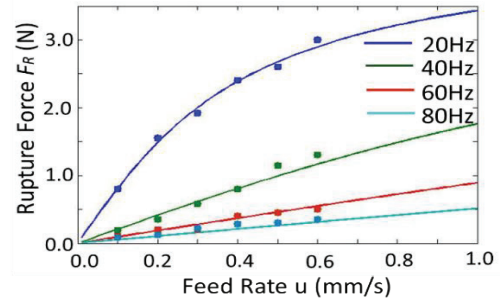


FIGURE 11. Measured rupture force versus different feed rates and cutting frequencies, plotted with the analytic model

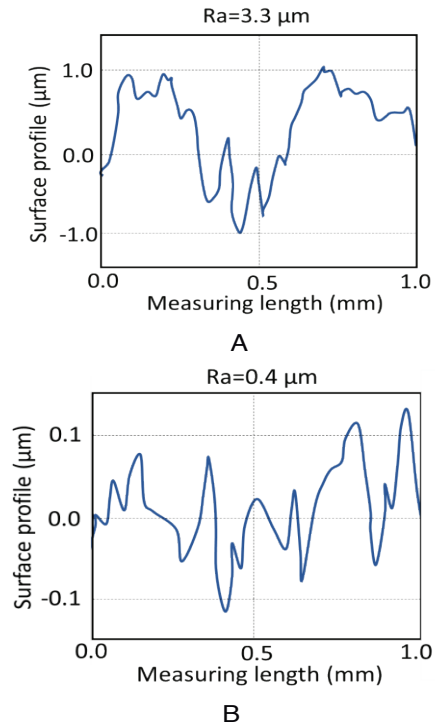


FIGURE 12. A: Surface profiles of the liver tissue cut at 20 Hz with a feed rate of 0.6 mm/s. B: Surface profiles of the liver tissue cut at 80 Hz with a feed rate of 0.1 mm/s

CONCLUSION

In this paper, we present a flexure-based vibrating blade microtome for ultra-precision tissue sectioning, achieving 10 μm thick tissue sections and surface roughness of 400 nm. Analytic models have been developed to describe the oscillatory cutting process based on fracture mechanics. Tissue sectioning experiments of different frequencies and feed rates were performed to validate the accuracy of the model. Results from the model and experiments confirm that low rupture force leads to high cut quality, which indirectly proves that the rupture force is a good measurement of section quality.

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REFERENCES

- [1] Chen S, Panas R, Culpepper ML, and Ragan T. Design of a Precision Flexure-Based Vibration Microtome for Whole Organ Imaging. Proc. of the Annual Meeting of the ASPE, San Diego, CA, Oct. 2012, pp.186-89.
- [2] Atkins T. The Science and Engineering of cutting: The Mechanics and Processes of Separating, Scratching and Puncturing Biomaterials, Metals and Non-Metals. Elsevier. Oxford. 2009.
- [3] Nestor Perez. Fracture Mechanics. Kluwer Academic Publishers. New York. 2004.
- [4] Fung, Y. C. Biomechanics: Mechanical Properties of Living Tissues. Springer. New York. 1993.
- [5] Mohsen Mahvash, Pierre E. Dupont. Mechanics of Dynamic Needle Insertion into a Biological Material. IEEE Transactions of biomedical engineering, Vol. 57, NO. 4, APRIL 2010.

DESIGN OF CONSTRAINT SYSTEMS BASED ON THE RAYLEIGH QUOTIENT AND MINIMAX PROPERTIES OF EIGENVALUES

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INTRODUCTION

Exact-constraint or kinematic design principles are central to precision machine design. A key step in many designs is the selection of a set of constraints that satisfy requirements on component tolerances, thermal mismatches, dynamics, and gravity sag. In this paper, we demonstrate the use of some properties of eigenvalues under constraint to quickly arrive at sets of constraints that maximize the resonant frequencies of structures based on their free resonant frequencies.

Numerous studies have investigated methods to optimize structures (see [Grandhi 1993] for a review paper). Hindle [1945] uses a whiffletree, sometimes referred to as a Hindle mount, to approximately minimize the deflection of a circular mirror under its weight. Yoder [2006] provides a figure comparing the optimal support radius for circular plates due to gravity with a number of supports. Nomura and Wang [1989] optimize the placement of supports under a free rectangular plate using a Galerkin method.

Wang et al [2004] develop closed form sensitivities and a heuristic method to optimize support locations for maximum natural frequency. Son and Kwak [Son and Kwak 1993] use eigenvalue sensitivities to optimize the boundary conditions and maximize the natural frequency of structures. Friswell [2005] uses finite elements which contain the support conditions within each element to optimize support placement and maximize natural frequency. Won and Park [1998] develop the concept of limit eigenvalue, derive conditions for which a support reaches the limit eigenvalue, and use optimal support loci to achieve it. Kesson and Olhoff [1988] use Courant's minimax principle to place spring supports of a beam in their optimal location to maximize its natural frequencies.

RAYLEIGH QUOTIENT AND THE EIGENVALUE MINIMAX PRINCIPLE

The theory of the Rayleigh quotient with constraints is summarized by Gladwell [2004] as follows:

"If a linear constraint is applied to a system, each natural frequency increases, but does not exceed the next natural frequency of the original system." Therefore, if the eigenvalues of a baseline system are written in sequence as $\lambda_1, \lambda_2, \lambda_3 \dots$ and the eigenvalues of the system obtained by adding a constraint are written as $\lambda'_1, \lambda'_2, \lambda'_3 \dots$, the new eigenvalues must obey

$$\lambda_1 \leq \lambda'_1 \leq \lambda_2, \quad \lambda_2 \leq \lambda'_2 \leq \lambda_3, \quad \lambda_3 \leq \lambda'_3 \leq \lambda_4$$

and so on, as illustrated in Fig. 1. From the Courant minimax principle, one further finds that a constraint which renders $\lambda'_1 = \lambda_2$ must exist. This is referred to as the *limit eigenvalue*.

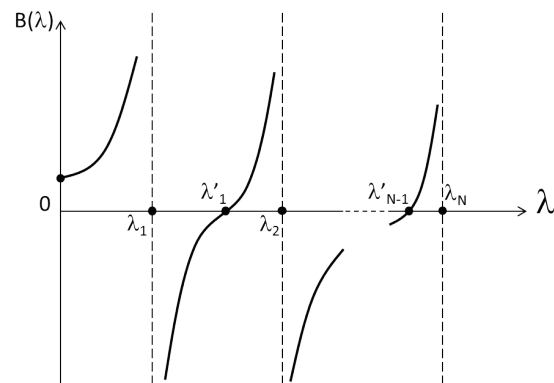


FIGURE 1. Change in natural frequency of an N degree of freedom system with one constraint added.

As an illustration, consider planar transverse vibration of a uniform beam. The first two free vibration modes are rigid-body modes, and the third vibration mode (first mode with non-zero frequency) is a symmetric bending mode with nodes located approximately 22% of the beam length from each end. If a constraint is added at each of these nodal points, then the rigid-body modes are eliminated and the first resonance of the constrained system is equal to the third natural frequency of the original (free) system. According to the Rayleigh quotient with constraints, this is the limit eigenvalue, and is the upper bound to which the first resonance can be raised with two constraints.

EXAMPLE: CIRCULAR PLATE

Suppose we wish to kinematically support a 300 mm silicon wafer. The first several free modes are plotted in Fig. 2. According to the mini-max principle, we should be able to find three constraints that render the first resonance of the constrained system equal to the fourth resonance of Fig. 2. But examining the repeated modes 4 and 5, we note that aside from the center, their nodal lines do not intersect. Therefore, we cannot simply fix three points under the plate and preserve both modes. In the following, we first determine the highest resonance that can be obtained with a three-point mount, then examine an alternate constraint set that does attain a first resonance equal to the fourth mode of Fig. 2.

Three-Point Mount

Consider first a three-point mount as sketched in Fig. 3. We can vary the radius r at which the three supports are placed and solve for the first resonance to attain the curve shown in Fig. 4. The highest attained resonance is 52 Hz and is obtained at $r/R \approx 0.4$. This is about 20% lower than the limit eigenvalue (the 4th free mode). The mode shapes attained with supports at this location are shown in Fig. 5.

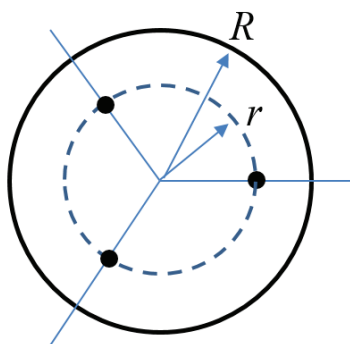


FIGURE 3. Sketch of a circular plate of radius R constrained at three points located on a circle of radius r and spaced at 120 degrees.

Though the optimum resonance of just over 52 Hz is obtained $r/R = 0.4$, one might in practice accept a somewhat lower resonance and place the supports at $r/R \approx 0.65$, which provides a broader support and lowers the peak-to-valley gravity sag in the wafer from about 90 to $44 \mu\text{m}$ while only dropping the resonance frequency from about 52 to 49 Hz.

Swingle Mount

Suppose instead that we choose the four points (A,B,C,D) shown in Fig. 6 and pick constraints on

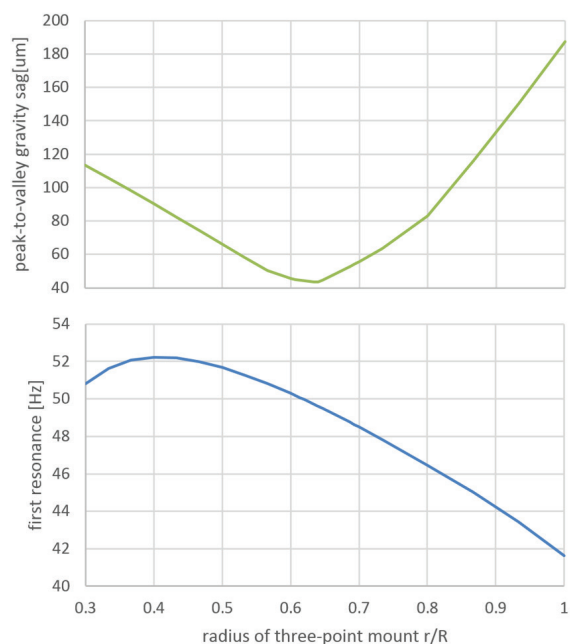


FIGURE 4. Plot of the first resonance frequency of the 300 mm wafer with three-point support as shown in Fig. 3 as a function of support location radius r . Also plotted above is the peak-to-valley gravity sag in the wafer.

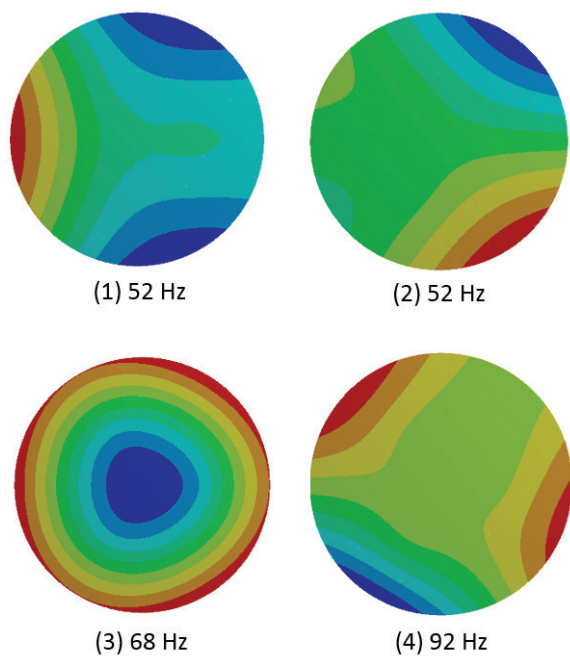


FIGURE 5. First four resonances of the 300 mm wafer with three-point support as shown in Fig. 3 with $r/R = 0.4$.

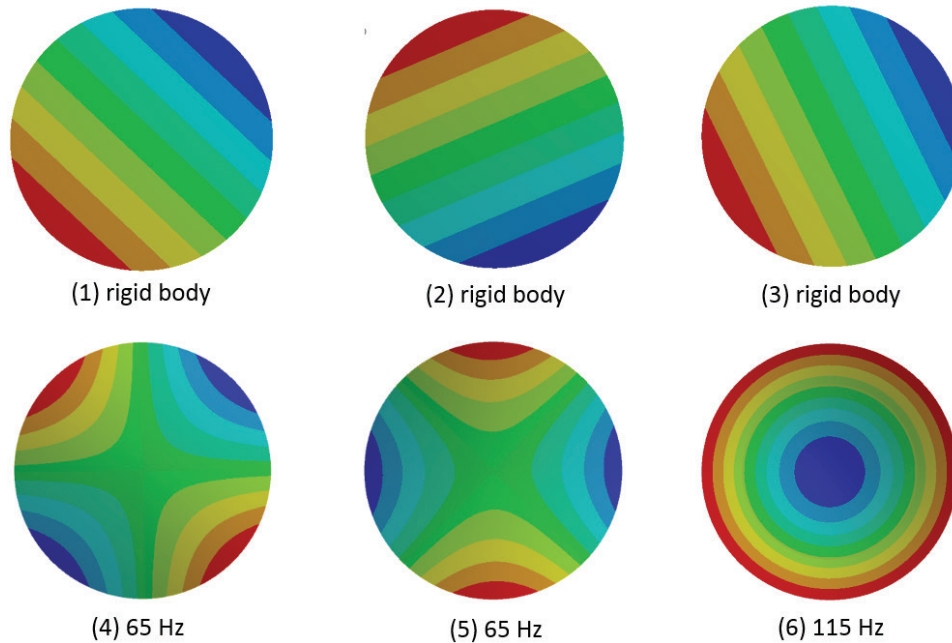


FIGURE 2. Color contours of the first six free modes of a silicon wafer 300 mm diameter and 0.775 mm thick. Silicon wafers are orthotropic, but for the purpose of illustrating the constraint method in this paper, we treat the silicon as isotropic with density 2300 kg/m³, elastic modulus 130 GPa, and Poisson's ratio 0.28.

the displacements w_A , w_B , w_C , and w_D in the form

$$w_A = -w_B, \quad w_B = -w_C, \quad \text{and} \quad w_C = -w_D \quad (1)$$

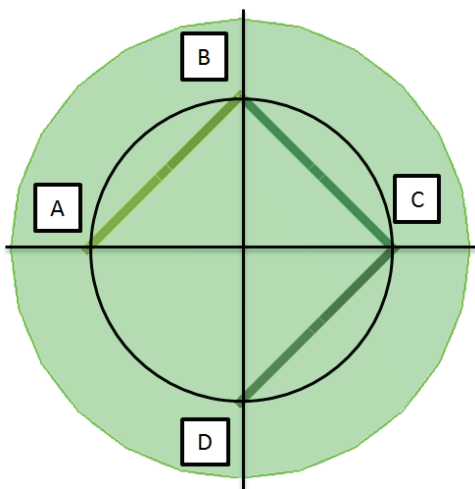


FIGURE 6. Top view of a silicon wafer indicating four points A, B, C, and D at which the wafer is supported. An isometric view of a linkage coupling the four points is shown in Fig. 7.

Each of these constraints corresponds to a swingle or whiffle as is commonly used to re-

duce gravity sag in kinematic mounts. A sketch of such a system is shown in Fig. 7, where the pivots are realized using notch flexures. A three-dimensional FEA analysis of the system as realized yields the modes shown in Fig. 8, from which we see that it is a good approximation to the constraint set of Eq. (1), and that such a constraint system does yield a first mode of the constrained wafer equal to the fourth mode of the unconstrained (free) wafer.

The condition given by Eq. 1 could be realized with A, B, C, and D at any radius from the center, and therefore modes (1) and (2) of Fig. 8 can be obtained with the supports at any radius. However, only for certain support radii will those modes have the lowest frequency. In this example, we have chosen to place the supports A, B, C, and D at the same radius as the nodal line of the 6th free mode at 115 Hz. This mode therefore appears unaltered as the 5th mode in the constrained system as shown in Fig. 8. The peak-to-valley gravity sag for this design is about 16 μm .

Modes (1), (2), and (5) in Fig. 8 involve no reaction to the base and therefore cannot be excited by vertical motion of the base. Therefore, for applications where base motion is the most impor-

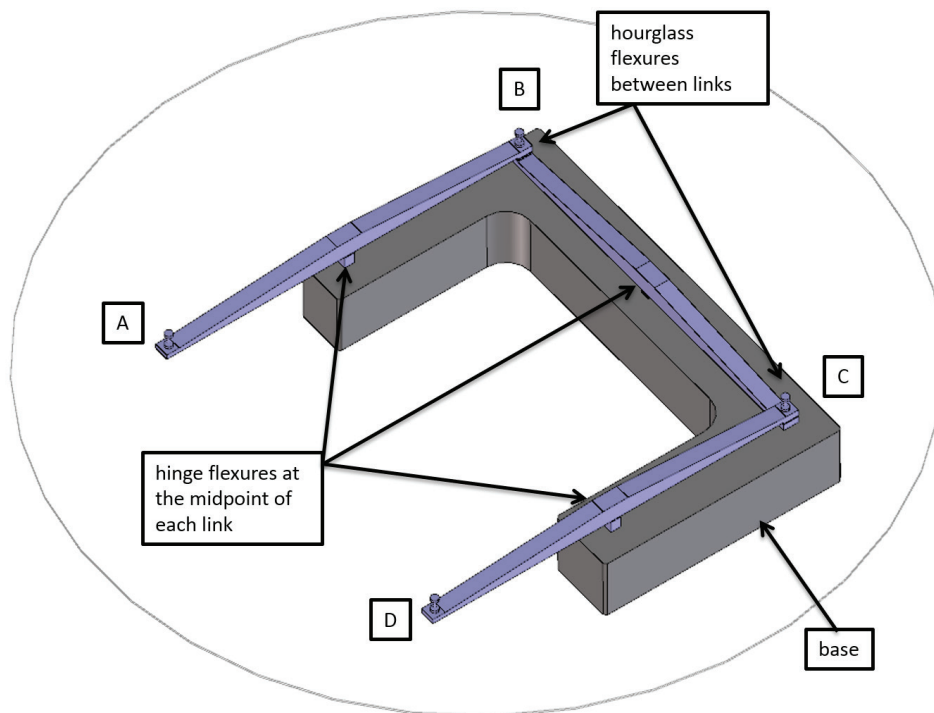


FIGURE 7. Concept for realization of the constraints of Eq. 1 in support of a 300 mm silicon wafer. In addition to the flexures labelled in the figure, at each of the contacts A, B, C, and D, there is an hourglass flexure to relieve in-plane stresses.

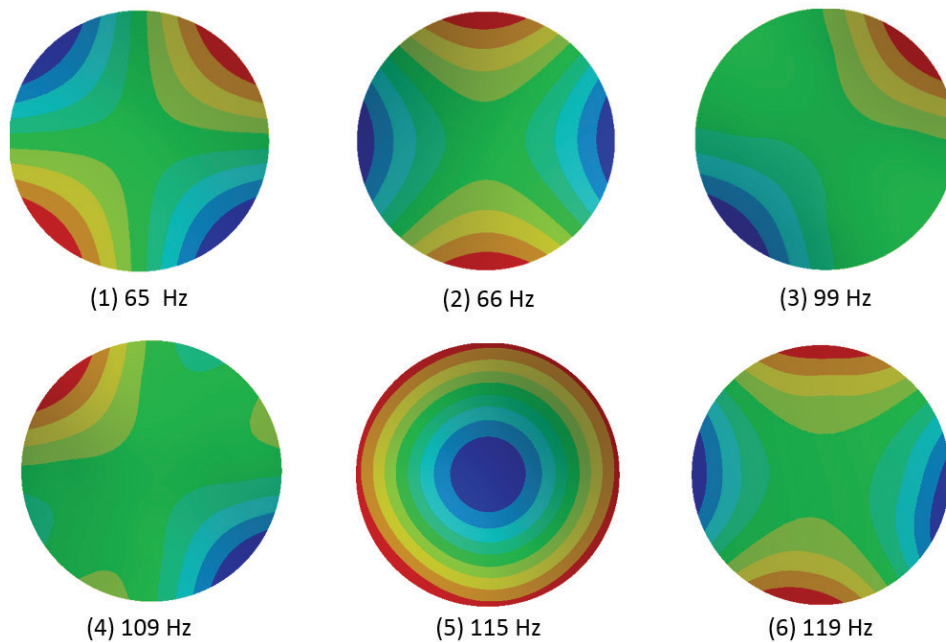


FIGURE 8. Mode shapes attained from a three-dimensional FE analysis of the design sketched in Fig. 7. The peak-to-valley gravity sag for this configuration is $16\ \mu\text{m}$.

tant disturbance, these modes become the focus of design effort. For the concept as sketched, they involve a significant amount of strain energy in the whiffle links, and could therefore be raised or lowered depending on the details of the link and flexure design. When considering such modes, we must think of the supports not as constraints, but as springs (and possibly masses).

For these higher modes of kinematically constrained systems (or for any modes of overconstrained systems) the compliance of the supports must be considered. Won and Park [1998] derive conditions for which the limit eigenvalue can be achieved when adding a support in overconstrained plates. In such cases, they show that the optimal location depends on the stiffness of the added support, and that above a critical stiffness, the limit eigenvalue is obtained by placing the support on a nodal line of the next higher mode of the system without added support.

CONCLUSION

The minimax properties of Rayleigh's quotient provide direct bounds on the changes in resonance frequencies that can be attained when constraints are added to a system. These mathematical properties map directly to the practice of exact-constraint design, and can guide the designer in the selection of constraints to maximize resonance frequencies. In cases where the minimax design requires whiffles, practical tradeoffs in the stiffness, inertia, and complexity of the design must be considered.

REFERENCES

- Friswell, M.I., 2006, "Efficient placement of rigid supports using finite element models." *Communications in Numerical Methods in Engineering*, 22:205–213.
- Gladwell, G.M.L., 2005, *Inverse problems in vibration*. Springer, Netherlands.
- Hindle, J.H., 1945, "Mechanical flotation of mirrors" *Advance Telescope Making*. Ingalls, A.G., ed. Scientific American, 229, New York.
- Son, Jae H. and Kwak B. M., 1993, "Optimization of boundary conditions for maximum fundamental frequency of vibrating structures", *AIAA Journal*, Vol. 31, No. 12, pp. 2351-2357.
- Kesson, B. and Olhoff, N., 1988, "Minimum stiffness of optimally located supports for maxi-

mum value of beam eigenfrequencies" *Journal of Sound and Vibration*, 120(3), 457–463.

Nomura, S. and Wang, B. P., 1989, "Free vibration of play by integral method." *Computers and Structures*. 32(1), pp. 245–247.

Grandhi, R., 1993, "Structural optimization with frequency constraints - A review", *AIAA Journal*, Vol. 31, No. 12, pp. 2296-2303.

Wang, D., Jiang, J. S. and Zhang, W. H., 2004, Optimization of support positions to maximize the fundamental frequency of structures. *Int. J. Numer. Meth. Engng.*, 61: 1584-1602.

Won, K.-M. and Y.-S. Park, 1998, "Optimal support positions for a structure to maximize its fundamental natural frequency" *Journal of Sound and Vibration*, 213(5), 801–812.

Yoder, Paul R. Jr., 2006, *Opto-Mechanical Systems Design*. CRC Press.

IS ONE STEP HEIGHT ENOUGH?

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INTRODUCTION

It is common practice in the field of surface topography measurement to determine the vertical amplitude response function of an instrument, tactile or otherwise, using a step height artefact, where the height of the step is known within a stated measurement uncertainty (i.e. has been calibrated) [1]. The measurement and subsequent data analysis techniques are well understood in the community, the analysis technique ensures stable, reproducible results and they are covered by international specification standards [2,3]. The use of step height artefacts has also become the *de facto* way to demonstrate that an instrument is “traceable”, to demonstrate that an instrument is capable of very high accuracy and the measurement results often form the “highlight” of the technical specification of a commercial instrument.

In this paper I want to argue that, whilst the amplitude response function of a surface topography instrument is a critical part of its specification, it is not enough to demonstrate the instrument’s ability to measure even moderately complex surface texture. More information about the instrument’s capability for measuring lateral features and surface slope angles should also be present in the instrument’s specification. Also, I will argue that *only* stating the ability of an instrument to measure step heights, should not be the industry standard and we should certainly expect more from the research community.

There is currently no standard way to present the technical specification of a surface topography instrument, although there are now efforts in the relevant ISO standards committee (ISO technical committee 213 working group 16) to address this lack [4] and the need has been discussed in the literature [5].

DETERMINATION OF THE VERTICAL METROLOGICAL CHARACTERISTICS

The amplification coefficient is defined as the slope of the linear fit to the output of a measuring instrument over a range of heights [1-3]. Figure 1 shows a typical example of a linear scale response curve. The linearity of the z-axis is given by the maximum deviation of the instrument response curve from the linear curve, where the amplification coefficient is given by the slope of the linear curve [6]. The z-axis calibration usually consists of a series of measurements of step height artefacts with various height values to establish the relationship between the ideal response curve and the instrument response curve. There have been attempts to integrate a displacement measuring interferometer with optical instruments (for example, [7]), but such methods can be costly and have not been adopted widely in industry. de Groot has recently proposed a method to calibrate the z-axis response of coherence scanning interferometers by effectively using the source as a traceable wavelength standard [8]. But, the step height remains the artefact of choice in industry and it is highly likely that it will become the default method when ISO specification standards on areal calibration are published (a default method is used in specification standards to denote the method that has been used if no method is specifically stated).

WHY THE VERTICAL METROLOGICAL CHARACTERISTICS ARE NOT ENOUGH?

There are countless examples (the author does not wish to cite examples, but they are easily found in the literature dating back many years) of published material which reports the development of new instruments for measuring surface topography, and the only performance data that is reported is a comparison of a step height measurement, usually using a step height with a certified height value (and hopefully associated measurement uncertainty). In many

cases only a very low number of height values are reported; sometimes just one. In some cases, it is then stated that the instrument in question is “calibrated” or “traceable” and it is assumed that it can measure surface texture. But how safe is this assumption?

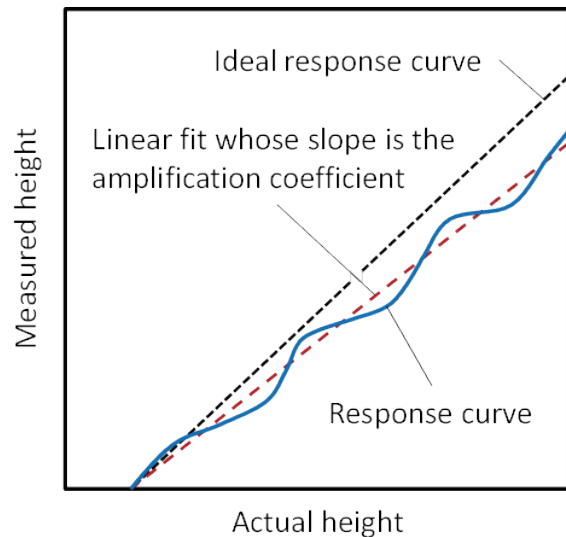


FIGURE 1. Illustrative response characteristic of a surface texture instrument

Firstly, surface topography instruments measure a discrete number of heights at (x,y) positions on a surface. Therefore, it is also important to determine the instrument response function in the x - y plane. For example, some parameters are defined in the x - rather than the z -direction such as the spacing parameter, RSm , or are a combination such as the RMS slope parameter, Sdq . Even when a parameter has only a z -component, such as Ra or a step height, the x - and y -coordinates may enter the value and its associated uncertainty indirectly *via* the filter definition [9].

Secondly, a surface topography instrument operates over a finite spatial frequency bandwidth. This is determined by such factors as the tip geometry and instrument range for a stylus instrument and the optical transfer function for an optical instrument (this is not an exhaustive list - these factors are discussed in detail elsewhere [10, 11]). Also, in many cases, the spatial frequency response of the instrument is not linear with height and/or spacing [12], implying that there can be both vertical and lateral distortion of the measured surface

features. A finite spatial frequency bandwidth also results in a finite slope range over which the instrument operates reliably. Simply calibrating the vertical and lateral scales of the instrument does not capture this bandwidth information. Therefore, when measuring surface texture, which contains features with varying geometry, it is important that the spatial frequency response of the instrument is understood. There is on-going work in ISO 213 WG 16 and the wider research community to try to design artefacts and methods to calibrate the (amplitude-dependent) spatial frequency response of an instrument (see for example, [13]), but there is no consensus yet reached.

Thirdly, noise and residual flatness affect any parameter, but this is only partially reflected in the repeatability of a step height value. Also, here the parameter direction is not decisive: noise in the z -axis may affect the RSm value, which is essentially an x -axis parameter.

Lastly, the probe or its optical equivalent may appear to measure a deformed surface, for example, a pure sinusoidal surface is deformed by a finite probe diameter or by an angular-dependent optical transfer function. Even if the wavelength transmission is only partially, or even not affected, the deformed measured surface can give rise to changed surface texture parameter values.

Note that when using only a single step, one cannot be sure whether one is determining the amplification coefficient or the deviation from linearity at a single point (or the one compensating the other when just the calibrated value is returned). Even when measuring a single step, one should do this in several areas of the z -axis range (if possible) in order to be able to conclude anything at all.

RESIDUAL FLATNESS

The amplitude response will not be representative of a surface measurement if the reference surface is not flat. The residual flatness will affect any measured parameter as a measurement is never about a single point on a surface. The residual flatness that can be obtained depends on the guideway and motion technology and range used, but it rarely approaches the sub-nanometre levels that are

sometimes quoted for the instrument repeatability.

VERTICAL RESOLUTION

The vertical resolution of surface topography instruments is almost always quoted on instrument specifications, although never with any uncertainty (to the author's knowledge). But what does the term mean? Does it have a similar meaning to lateral resolution, *i.e.* the ability to distinguish two features unambiguously? If it does, then how is it determined? It is difficult to manufacture an appropriate artefact, so in practice step height vendors quote vertical resolutions of less than a nanometre, sometimes orders of magnitude less (again, the author does not wish to cite examples). Whilst sub-nanometre step height artefacts do exist (see for example [14]), they are difficult to use in practice, have a tendency for secular dimensional drift and have uncertainties that are of the same order of magnitude as the measurand. Some vendors simply quote noise values (usually over a relatively large field of view and in terms of S_q) and I have already discussed the problems with this; others simply rely on theory to come up with a number. This practice has become somewhat alarming over the last decade – a competition to quote the smallest number; justified as a necessary marketing ploy. However, to be fair to vendors, when one's livelihood depends on sales as opposed to government grants, and the process to develop standards can take a very long time, the outcome is predictable.

POSSIBLE WAYS FORWARD

ISO TC 213 WG 16 is currently developing a framework for calibration and performance verification of surface topography instruments. Current instrument standards (the "6XX Series [4]) include a table that attempts to list the majority of the factors that can affect the uncertainty of a measurement with the specific instrument. Clearly, such a list can never be exhaustive, can never take into account all measurement scenarios and it is a tall order to ask the average instrument user to determine all these influence factors. For this reason, WG 16 developed the "metrological characteristics",

formally defined in ISO 25178 part 600 as "characteristics of a measuring equipment, which may influence the results of measurement" (this definition is not published yet but is unlikely to change significantly). Metrological characteristics are an attempt to reduce the calibration of an instrument and hence, any subsequent estimation of measurement uncertainties using that instrument, to the experimental determination of a manageable number of parameters using calibrated material measures. The current list is given below and is described in detail elsewhere [6,15,16] (with the exception of the topography fidelity, which is still under discussion [13]):

- Amplification coefficient.
- Linearity deviation.
- Residual flatness.
- Measurement noise.
- Topographic spatial resolution.
- Perpendicularity.
- Topography fidelity.

Note that it is highly likely that a metrological characteristic will be added which accounts for the finite slope measuring capability of an instrument (for example, the maximum measurable slope). Once the metrological characteristics have been determined, they can be used as the quantitative values in a measurement model to determine task-specific measurement uncertainties. Clearly, not all metrological characteristics apply to all measurement tasks, for example, if a step height is being measured, there is little point in considering the lateral period limit, so long as the step width is a lot larger than the lateral resolution of the instrument (and they usually are).

ISO 25178 part 700 will describe how to determine the metrological characteristics, but is likely to take another five years before publication. Hopefully, part 700 will also address performance verification and maybe even instrument specification, but this is still open to debate.

As suggested in reference [5], why can we not base our instrument specifications on a standard set of experimentally determined metrological characteristics? This would avoid confusion and allow potential users to directly compare instrument performance. However, once we

have the instrument vendors using a standard specification, there is another hurdle to leap. We then need to get the instrument development and research community to start using the standardised list of specifications in their publications. It is hoped that this paper can be used to encourage researchers to show the results of more than just step height measurements.

The use of a single step height to show instrument performance has had its day. It is hoped that the ISO specification standards under development will allow researchers, instrument developers, vendors and users to use a common framework for instrument specification, calibration and performance verification.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] Leach R. K. The Measurement of Surface Texture Using Stylus Instruments. 2nd Edition, Good Practice Guide 37, National Physical Laboratory: 2014.
- [2] ISO 5436-1 Geometrical Product Specification (GPS) – Surface Texture: Profile Method – Terms, Definitions and Surface Texture Parameters. International Organization of Standardization. 2000.
- [3] ISO WD 25178-600 Geometrical Product Specifications (GPS) — Surface Texture: Areal — Part 600: Metrological Characteristics for Areal-Topography Measuring Methods. International Organization for Standardization. 2015.
- [4] Leach R. K., de Groot P. J. The Standards Rash: Is there a Cure? *Int. J. Metrol. Qual. Eng.* 2015; 6:101.
- [5] de Groot P. J. Progress in the Specification of Optical Instruments for the Measurement of Surface Form and Texture. *Proc. SPIE.* 2014; 9110:91100M.
- [6] Giusca C. L., Leach R. K., Helery, F. Calibration of the Scales of Areal Surface Topography Measuring Instruments: Part 2. Amplification, Linearity and Squareness. *Meas. Sci. Technol.* 2012; 23:065005.
- [7] Kiyono S., Gao W., Zhang S., Aramaki T. Self-Calibration of a Scanning White Light Interference Microscope. *Optical Engineering.* 2000; 39:2720.
- [8] de Groot P. J., Beverage J. Calibration For The Amplitude Scaling Coefficient In Interference Microscopy by Means of a Wavelength Standard. *Proc. SPIE.* 2015; 9526:37.
- [9] Haitjema H., Uncertainty Estimation of 2.5-D Roughness Parameters Obtained by Mechanical Probing. *Int. J. Prec. Techn.* 2013; 3:403.
- [10] de Groot P., Colonna de Lega X. Interpreting Interferometric Height Measurements Using the Instrument Transfer Function. *Fringe 2005.* Springer Berlin Heidelberg, 2006; 30-37.
- [11] Coupland J., Mandal R., Polodhi K., Leach R. K. Coherence Scanning Interferometry: Linear Theory of Surface Measurement. *Appl. Opt.* 2013; 52:3662.
- [12] Colonna De Lega X., de Groot P. Lateral Resolution And Instrument Transfer Function As Criteria For Selecting Surface Metrology Instruments. *Optical Fabrication and Testing, OSA.* 2012.
- [13] Seewig J., Eifler M., Wiora G. Unambiguous Evaluation of a Chirp Measurement Standard. *Surf. Topog.: Metrol. Prop.* 2014; 2:045003.
- [14] Orjia N. G., Dixon R. G., Fub J., Vorburger T. V. Traceable Pico-Meter Level Step Height Metrology. *Wear.* 2004; 257:1264.
- [15] Giusca C. L., Leach R. K., Helery F., Gutauskas T., Nimishakavi L. Calibration of the Scales of Areal Surface Topography Measuring Instruments: Part 1 - Measurement Noise and Residual Flatness. *Meas. Sci. Technol.*, 2012; 23:035008.
- [16] Giusca C. L., Leach R. K. Calibration of The Scales of Areal Surface Topography Measuring Instruments: Part 3 - Resolution. *Meas. Sci. Technol.*, 2013; 24:105010.
- [17] ISO CD 25178-700 Geometrical Product Specifications (GPS) — Surface Texture: Areal — Part 700: Calibration and Verification of Areal-Topography Measuring Methods. International Organization for Standardization. 2015.

Three Dimensional Digital Holographic Aperture Synthesis for Rapid and Highly Accurate Large Volume Metrology

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INTRODUCTION

Currently large volume, high accuracy three-dimensional (3D) metrology is dominated by laser trackers [1], which typically utilize a 2D galvanometer and a time-of-flight (TOF) lidar system (base unit) along with a cooperative reflector to estimate points on a given surface. The general mode of operation for a laser tracker is to have the base station track the cooperative target as it is moved along the surface of interest. This process typically requires two operators, one moving the cooperative target and one at the base station. During this process the base station measures the 3D coordinates of the cooperative target and these are recorded for later use. While the measurement times of the actual TOF lidar system can be fast, the combination of dual operators and the need for a cooperative target, make the overall application to large volume metrology very time consuming. Typically operators are limited to using the system at well-defined points known as key characteristics (KCs) on large structures such as airplane fuselages. The measurement of such KC points for the F-35 Joint Strike Fighter is a well-known bottleneck in the manufacturing process [2].

To increase speed, laser scanners or structured illumination systems can be used directly on the surface of interest. For example, laser scanners typically collimate a laser beam and use a combination of either scan mirrors or rotary stages to scan the laser across the surface of interest. A lidar system is then used to measure the 3D coordinates to the target and the surface is reconstructed. Such systems built for large volume metrology claim accuracies of around 250 μm at 25 meter standoff ranges with measurement rates of >1000 points per second [3]. It is important to note that both laser scanners and structured illumination systems are restricted in their lateral resolution (and

depending upon the type of system the axial resolution) at longer stand-off distances due to the diffraction limit of the objective used. To truly achieve a Rayleigh limited lateral resolution of 250 μm at 25 m, the transmit and receive objective would require at least a 20 cm diameter for a 1.5 μm wavelength. Most laser scanner vendors utilize objectives with 2.5 to 5 cm diameters, as larger optics and the required scanning systems become cost prohibitive. For these smaller sized objectives, the lateral resolution is typically on the order of 1 to 2 mm.

Synthetic aperture lidar (SAL) [4], distributed or sparse aperture imaging [5], and holographic aperture lidar (HAL) [6],[7], are coherent imaging techniques that coherently combine spatially and temporally diverse target returns to overcome the conventional diffraction limit. By recording individual estimates of the electric field with either a single receiver or a sensor array and by considering the relative motion between the transceiver and the target, these field estimates can be registered or "stitched" in the pupil plane to produce enhanced resolution two dimensional (2D) imagery. In order to form 3D imagery, two or more baselines in an interferometric SAL system [8], or two or more wavelengths in a distributed aperture or HAL system [9] have been proposed and demonstrated. The unresolved dimension is then derived from the measured phase difference between the spatial or wavelength multiplexed 2D images. Because these 3D imaging techniques rely on a relatively few number of synthetic wavelengths or spatial frequencies, targets with significant structure or discontinuities are poorly rendered.

In their paper, Duncan and Dierking briefly suggest that the HAL approach, when paired with a fully coherent ranging system, could overcome this limitation and provide fully

resolved 3D imagery while preserving the ability to enhance the lateral resolution via aperture synthesis [6]. In this paper, we demonstrate fully down range resolved coherent imaging with aperture synthesis by use of a FMCW chirped heterodyne lidar system coupled with a fast focal plane array. We term the approach 3D-HAL for brevity throughout this paper because the approach derives much of its aperture synthesis techniques from Ref. [6]. However, the approach can also be considered as extending the 3D imaging approach described by Marron by applying aperture synthesis techniques in the lateral dimensions [10]. Recently, Stafford et. al published a very short report on a demonstration of the technique when operated in the inverse synthetic mode [11].

Because 3D-HAL relies upon a parallel readout architecture, it has advantages similar to direct-detect flash lidar systems. However, because the approach is coherent, it has many additional capabilities including: 1) enhanced lateral resolution through aperture synthesis approaches, 2) access to ultra-high axial or down range resolutions through stretched processing of chirped waveforms, 3) the ability to utilize digital phase correction and image sharpness metrics for correcting optical or atmospheric aberrations [7], and 4) advanced Doppler or interferometric processing to measure the coherent evolution of the 3D point cloud. The 3D-HAL approach also uses heterodyne detection, which is extremely sensitive when shot-noise-limited, and which enables coherent integration to increase carrier-to-noise ratios. This makes 3D-HAL ideally suited for use in photon-starved environments such as large volume metrology. In this paper, we provide background for the 3D-HAL approach and provide simulation and experimental results for 3D aperture synthesis. We also explore the coherent nature of the point cloud, by tracking the phase evolution of 3D images when the target undergoes thermal expansion, allowing interferometric scale point-cloud resolution.

BACKGROUND

HAL is a form of digital holography that resolves both transverse dimensions of the scene, but not the downrange dimension [6],[7]. Reference [5] provides a thorough theoretical treatment of aperture synthesis using a sensor array including both spotlight and stripmap holographic transformation and the reader is

referred there for details of the approach. Through the use of these transformations, the pupil-plane field segments are properly registered so that a larger synthesized complex pupil-plane field is created without residual errors due to the given translational or rotational geometry. The sensor array can be placed in the pupil plane or in the focal plane, since they are connected through a Fourier transform.

In 3D-HAL, we introduce a temporally-modulated, linear-chirp waveform for ranging at each aperture location, which is similar to a SAL system. While any coherent form of laser ranging could be utilized, FMCW chirped heterodyne ranging has the advantage of providing very high downrange resolution with a reduced receiver bandwidth via optical stretched processing [12]. The range resolution, δ_R , of this technique is limited by the chirp bandwidth, B , and is described by the relation, $\delta_R = c/2B$, where c is the speed of light.

FIGURE 1(Left) shows a schematic diagram for a generalized 3D-HAL system. Here a chirped transmit source, after being suitably amplified, flood illuminates a target. A fast FPA is used to record both the temporal and spatial interference between a local oscillator (not shown) and the complex return field for a given location along the dimension of motion. This is referred to as a field segment, $g_o(x, y, t_f)$, where x and y are lateral coordinates and t_f is the time during which the temporal interference is recorded (i.e. “fast time” in SAL formalism). Throughout this paper, we refer to sensor arrays that are capable of recording this temporal interference as “fast FPAs” to distinguish them from more traditional sensor arrays that operate with very slow frame rates. However, in 3D-HAL the fast FPAs need not be restricted to being placed in the “focal plane” as the acronym suggests.

Aperture synthesis depends on relative translation between the speckle field and the receive aperture in the aperture plane. A variety of configurations achieve this effect [4-8]. The locations for the transceiver are labeled alphabetically a-g in FIGURE 1 (Left) and we assume that the transmitter moves with the fast FPA, which is not strictly necessary [5]. The synthetic aperture size, d_{SA} , is shown and the enhanced lateral resolution is nominally $\delta_{CR} = \lambda R / (2d_{SA} + d_{AP})$, where λ is the carrier wavelength, R is the range to target, d_{AP} is the

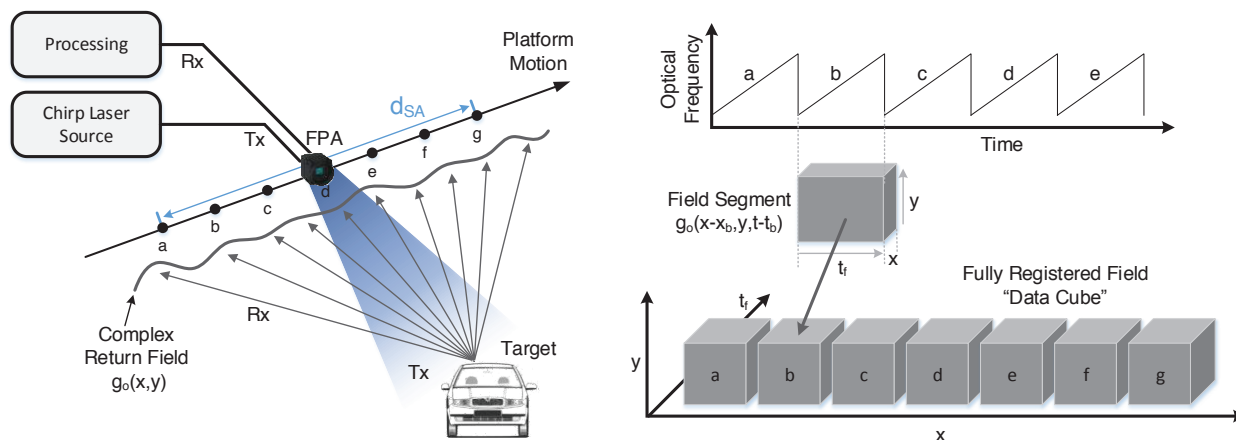


FIGURE 1. Schematic of a generalized 3D-HAL system. See text for description.

focal plane array width in the dimension of travel, the step size is assumed to be $d_{SA}/2$ and unity optical magnification is assumed [6].

FIGURE 1 (Right) shows the corresponding sequence of chirp waveforms and an example recorded field segment for aperture location “b” is drawn as a small cube. It is important to note that the fast FPA samples the field spatially at the pixel spacing, while the frame rate of the fast FPA gives the temporal sampling. The pixel spacing sets the overall field-of-view of the system in the lateral dimensions, while the frame rate is proportional to the maximum range difference that can be measured in the downrange dimension. Each recorded field segment is subsequently registered during a post-processing step with the other field segments, resulting in a larger synthesized or fully registered field estimate as shown in the bottom of FIGURE 1 (Right). We also refer to this structure as a “data cube”. Registration can either be performed by measuring the aperture positions with high precision or by using data driven registration techniques with sufficient overlap between field segments. Even after registration, the field segments typically remain incoherent with each other because the aperture motion must technically be compensated to less than the optical carrier wavelength, which is often not physically possible. Phase gradient autofocus or prominent point phasing algorithms are used to achieve coherence across all the field segments and the data can be further enhanced using digital image correction for atmospheric or other aberrations [7].

This data cube can be processed in a variety of ways to form useful 2D and 3D imagery. For example, any 2D slice through both transverse

dimensions for a fixed fast time can be processed as a HAL image using a 2D fast Fourier transform (FFT). Similarly, any 2D slice of the data cube in a downrange / transverse combination can be processed as a SAL image again using a 2D FFT. However, by transforming the whole data cube, with a 3D FFT, all dimensions are compressed into a fully resolved 3D distribution. Once fully compressed, peak extraction is performed along the downrange dimension to construct a point cloud.

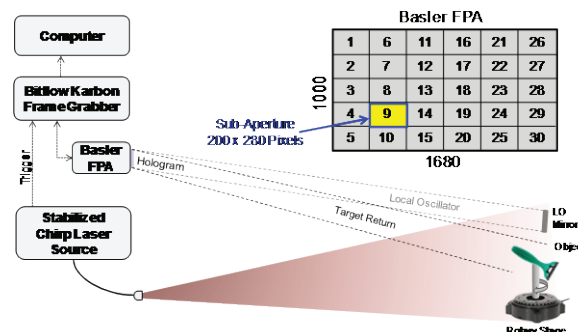


FIGURE 2. System schematic. Note the size of the field segment, colored in yellow, relative to the full FPA. This is the basis of our demonstrated cross range resolution enhancement.

EXPERIMENTAL DEMONSTRATIONS

3D-HAL demonstrations were conducted using a lens-less digital holographic setup shown in FIGURE 2. A stabilized FMCW linear-chirp laser at 1064 nm was used to flood illuminate various targets 2.5 meters away, each of which was mounted on a rotary stage. The laser output power was approximately 15 mW. The linear chirp had a bandwidth of 102 GHz and was repeated at a 5 Hz rate. A mirror near the

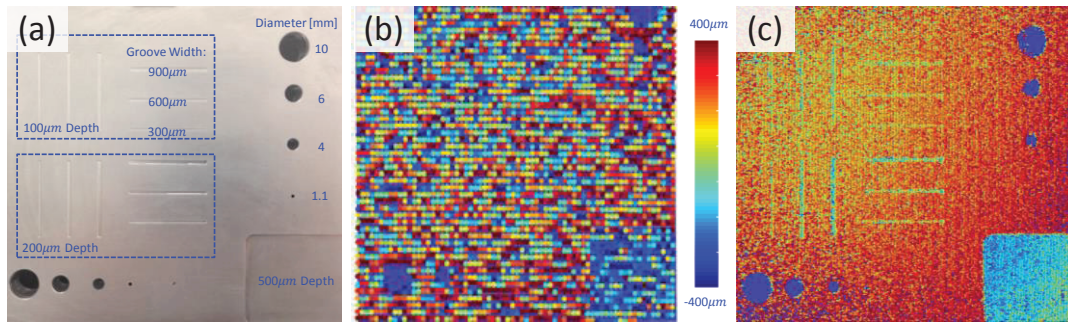


FIGURE 3. Demonstration of resolution enhancement using 3D-HAL. (a) Photograph of an aluminum plate overlaid with machined features. (b) Point cloud derived from processing of a single 200x280 field segment. (c) Point cloud derived from processing of the fully synthesized aperture of size 1000x1960 colored by depth out of the plane.

target was used to create a local oscillator beam. A Basler acA2000-340km CMOS array recorded the interference between the local oscillator and the target returns synchronized with the chirp output. A Bitflow Karbon frame grabber card was used to capture a 200x280 pixel region-of-interest (ROI) at a frame rate of 1.8 kHz. The resulting field segments had 360 frames in fast time and were 200 ms in duration.

The ROI was then shifted on the camera in order to sample the full spatial extent of the fast FPA and to provide Fourier fill for aperture synthesis. This is effectively a bi-static mode with a moving receive aperture and stationary transmitter. The bi-static mode halves the synthetic aperture and doubles the resolution [5], [8]. However, the stationary fast FPA results

in ideal spatial placement of the field segments and simplifies the processing. The size of the ROI limits the fast FPA to a 5x7 tiling resulting in 1000 x 1960 utilized pixels, similar to that shown in FIGURE 2 (Top Right). Given the PRF, the total collection time of a complete data cube is roughly 7 seconds. Several data cubes can be collected with no other changes to allow for coherent integration of the time domain signal for increased SNR. Alternatively, the rotary stage could be used to shift the target slightly resulting in a data cube with an independent speckle realization for speckle averaging. In the data presented below, four speckle realizations for the full data cube were averaged to reduce speckle effects.

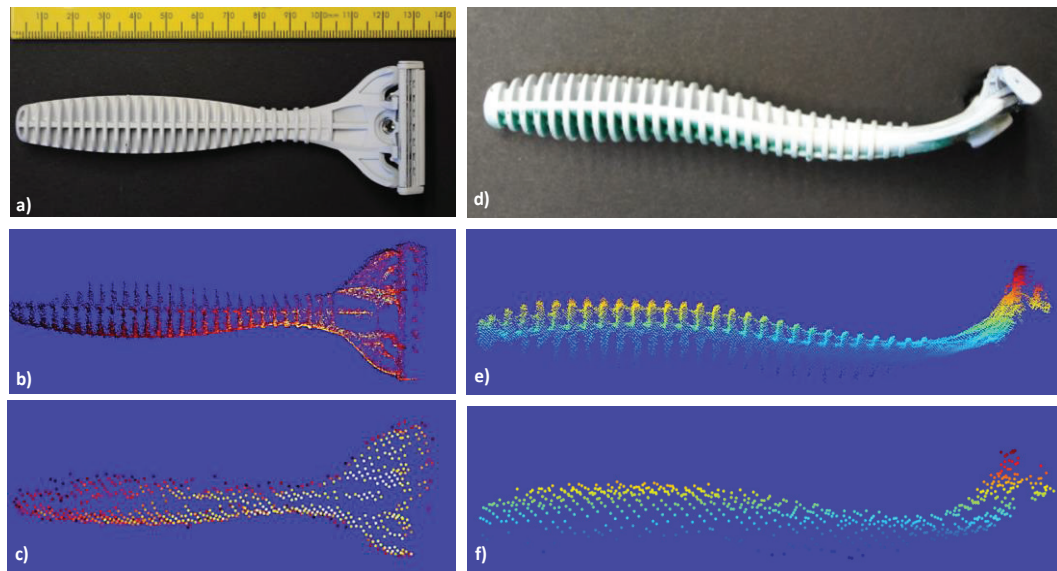


FIGURE 4. (a) and (d) Photographs of a shaving razor. (c) and (f) Single field segment and (b) and (e) full synthesized aperture 3D-HAL renderings of the razor showing significant resolution enhancement.

To demonstrate resolution enhancement, we first imaged a satin finished aluminum plate with precision machined features. The features included vertical and horizontal grooves at three different widths (300, 600, and 900 μm) and two different depths (100 and 200 μm). A corner of the aluminum plate was removed at a depth of 500 μm to assess the “step” resolution of the system. Holes of varying diameters (1, 4, 6 and 10 mm) were also drilled through the plate. The 3D image was formed using a single field segment as well as the entire coherently stitched array to demonstrate the resolution enhancement capability. The results of this experiment are shown in FIGURE 3.

To demonstrate resolution enhancement, we imaged a common shaving razor. FIGURE 4 (a and d) show photographs of the razor from different viewpoints. The razor was painted white to increase the return light; however with more illumination power this would not have been necessary. The razor was imaged with the 3D-HAL system and FIGURE 4 (c and f) show the resulting 3D point cloud from a single field segment. The rough shape of the razor is visible and the voxel size is estimated to be around 1-2 mm. FIGURE 4 (b and e) show the fully synthesized aperture 3D-HAL imagery revealing significantly more detail as expected. In this case, the system resolves the grooves in the handle and near the razor head. The voxel size is estimated to be around 100-200 microns. The point cloud renderings used a log scale intensity shading and the various viewpoints are from the same point cloud (i.e. the original image was taken from only one illumination direction). Clearly, the enhanced resolution provides significantly more information about the targets shape, location and features. In a practical metrology system, the sub-apertures

could be created by translating a small transceiver using a precise mechanical stage. This would reduce the cost while increasing the performance of future large volume metrology.

Finally, to fully demonstrate the coherent nature of point clouds rendered with the 3D-HAL approach, the phase information of the various points was tracked over sequential captures. Because this phase is proportional to the wavelength of the optical carrier, very small scale changes can be observed and mapped to the 3D rendering. To explore this capability, we imaged the thermal expansion of a coffee cup in the seconds after hot water was introduced into the container. The ROI was 400x560 pixels, with 2x2 binning, resulting in a narrower field-of-view, which allowed full data cubes to be captured at a 5 Hz frame rate. The phase value of each voxel over the sequential capture was then referenced to the phase of the identical voxel in the first capture. This phase was unwrapped and rendered onto the point cloud surface reconstruction to visualize thermal expansion of the mug on the micron scale. The results are shown in FIGURE 5. The observed phase evolution demonstrates that more thermal expansion is occurring in the bottom half of the mug where the hot water is present. In order to correctly utilize this phase to measure displacement, the surface normal must be estimated and compared to the line of sight of the 3D-HAL transceiver.

A potential metrology application of the demonstrated volumetric change detection technique is high resolution part inspection under thermal changes. The part could be conductively heated or place in a temperature controlled environment. Such an approach would be a powerful tool for thermal model

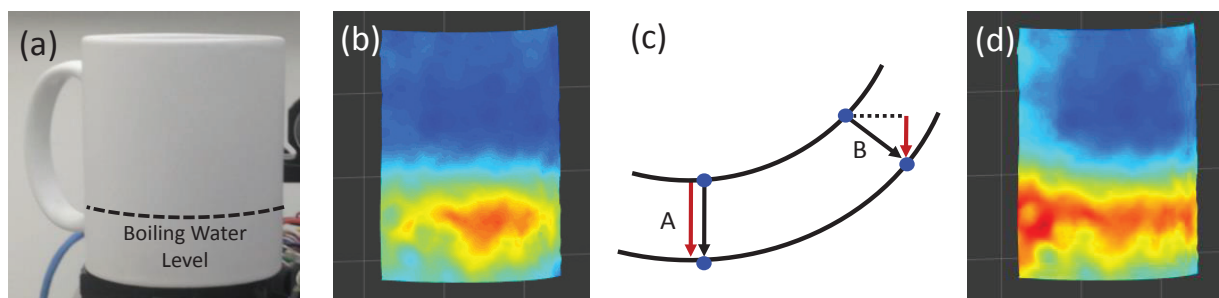


FIGURE 5. (a) Picture of the coffee mug and the boiling water level. (b) A colormap of the unwrapped phase after 60 seconds rendered onto the point cloud. (c) The observed phase change is proportional to the red arrows, which represent the dot product of the thermal expansion along the line of sight of the system. (d) By taking this projection into account, an estimate of the displacement can be made.

verification and could also prove useful in uncovering minute part defects or material inhomogeneity.

CONCLUSIONS

In this paper, the 3D-HAL technique has been introduced and demonstrated. Aperture synthesis using stitched field segments showed dramatic resolution enhancement over single field segments while fully resolving the 3D information content of the target. The fully coherent nature of the point clouds was also demonstrated by tracking the phase of a thermally expanding coffee cup. Metrology applications for this technique were discussed.

Because 3D-HAL uses FPAs which have a parallel readout, it is similar to direct-detect flash lidar systems. However, the approach relies on coherent detection and therefore has many advantages over direct-detect systems. While the approach is similar to both HAL and SAL architectures, the use of coherent downrange waveforms provides unprecedented extraction of 3D content. These 3D images are fully resolved and will perform better at 3D imaging targets with significant structure or discontinuities when compared to multi-wavelength HAL or multi-baseline SAL approaches. There are many possibilities for further research for 3D-HAL including exploration of larger synthetic apertures, data-driven registration of the field segments, more advanced Doppler processing of moving targets, and exploring longer range performance.

REFERENCES

- [1] D. Huo, P. G. Maropoulos, C. H. Cheng, "The Framework of the Virtual Laser Tracker – A Systematic Approach to the Assessment of Error sources and Uncertainty in Laser Tracker Measurement," Proceedings of the 6th CIRP-Sponsored International Conference on Digital Enterprise Technology Advances in Intelligent and Soft Computing Volume 66, 2010, pp 507-523.
- [2] See for example Air Force SBIR topic 112-119 entitled "Key Characteristics Metrology Solution," at <http://www.acq.osd.mil/osbp/sbir/solicitations/sbir20112/af112.htm>.
- [3] http://www.nikonmetrology.com/en_US/Products/Large-Volume-Applications/Laser-Radar/MV330-MV350-Laser-Radar/%28specifications%29
- [4] S. M. Beck, J. R. Buck, W. F. Buell, R. P. Dickinson, D. A. Kozlowski, N. J. Marechal, and T. J. Wright, "Synthetic-aperture imaging laser radar: laboratory demonstration and signal processing," Appl. Opt. 44, 7621–7629 (2005)
- [5] D. J. Rabb, D. F. Jameson, J. W. Stafford, and A. J. Stokes, "Multi-transmitter aperture synthesis," Opt. Express 18, 24937–24945 (2010)
- [6] B. D. Duncan and M. P. Dierking, "Holographic aperture lidar," Appl. Opt. 48, 1168–1177 (2009)
- [7] A. E. Tippie, A. Kumar and J. R. Fienup, "High-resolution synthetic-aperture digital holography with digital phase and pupil correction," Optics Express 19 (13), 12027–12038 (2011)
- [8] S. Crouch and Z. W. Barber, "Laboratory demonstrations of interferometric and spotlight synthetic aperture lidar techniques," Opt. Express 20, 24237–24246 (2012)
- [9] B. R. Dapore, D. J. Rabb, and J. W. Haus, "Phase noise analysis of two wavelength coherent imaging system," Opt. Express 21, 30642-30652 (2013)
- [10] Joseph C. Marron and Kirk S. Schroeder, "Three-dimensional lensless imaging using laser frequency diversity," Appl. Opt. 31, 255-262 (1992)
- [11] J. Stafford, B. Duncan, and D. J. Rabb, "Range-compressed Holographic Aperture Lidar," in *Imaging and Applied Optics 2015*, OSA Technical Digest (online) (Optical Society of America, 2015), paper LM1F.3.
- [12] P. A. Roos, R. R. Reibel, T. Berg, B. Kaylor, Z. W. Barber, and W. R. Babbitt, "Ultrabroadband optical chirp linearization for precision metrology applications," Opt. Lett. 34, 3692–3694 (2009).
- [13] Gatt, D. Jacob, B. Bradford, J. Marron and Brian Krause, "Performance bounds of the phase gradient autofocus algorithm for synthetic aperture lidar," Proc. SPIE 7323, Laser Radar Technology and Applications XIV, 73230P (2009)

SYSTEMS-LEVEL INTEGRATION FOR FOCUSED BEAM SCATTEROMETRY

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INTRODUCTION

The state of the art in optical lithography continues to push transistor critical dimensions below 15 nm, creating the need for extremely high-resolution metrology techniques as a result [1,2]. The International Technology Roadmap for Semiconductors has projected that the transistor logic half-pitch (critical dimension) reach 10 nm in industry by 2025 (See Table 1 below). Towards this end, the metrology of such fine features is a critical enabling technology. Ideally, the inspection of nanoscale features would be performed on an *a priori* basis; in reality, however, the features are well beyond the diffraction limit of the light used to inspect them, creating the need for more advanced strategies with longer inspection times.

Optical scatterometry and ellipsometry have evolved as some of the leading techniques for nano-scale feature inspection. One primary advantage of such methodologies is that they are significantly less expensive compared to well established alternatives (scanning electron microscopy, atomic force microscopy, scanning tunneling microscopy, x-ray inspection). Scatterometry and ellipsometry describe a collection of methods used to analyze the diffracted

spectrum (typically in the far field) from periodic features created during the wafer fabrication process. Unknown feature errors are determined by comparing measurements and rigorous simulations. Our unique adaptation, which we call Focused Beam Scatterometry (FBS) [3], aims to combine the strengths from a variety of the leading solutions in this field.

Some of the most reliable methods in industry include normal incidence scatterometry, spectroscopic ellipsometry, and 2- θ scatterometry [4]. Normal-incidence scatterometry is the simplest and involves a coherent or incoherent source that is incident on the inspected sample in the normal direction. Spectroscopic ellipsometry identifies scattered light as a function of wavelength and polarization. Lastly, 2- θ scatterometry operates by measuring scattered light as a function of the angle of the incident light.

WORKING PRINCIPLE

The working principle of our approach is as follows. First we consider a sample with periodic nano-scale features with expected nominal dimensions (critical dimension, sidewall angle, feature height, oxide layer thickness). Second we create a simulation

TABLE 1. Summary projections of the ITRS technology trend targets [1]

Year of Production	2013	2015	2017	2019	2021	2023	2025	2028
Logic $\frac{1}{2}$ Pitch [nm]	40	32	25	20	16	13	10	7
Flash $\frac{1}{2}$ Pitch (2D) [nm]	18	15	13	11	9	8	8	8
DRAM $\frac{1}{2}$ Pitch [nm]	28	24	20	17	14	12	10	7.7

of what the scattered light would look like after reflecting off the sample. In FBS we analyze the spatially varying polarization of the scattered light. After creating our initial simulation, we then explore a range of errors which could possibly occur. The simulations are carried out using Rigorous Coupled Wave Analysis (RCWA) [5,6] which connects the incident direction vector with all transmitted and reflected orders of the inspected grating. A library is created for slight errors, for example, one simulation may predict the output for all accurate dimensions except for a sidewall angle error of 0.10 degrees. In the next simulation, all dimensions are held constant except the sidewall angle error is changed to 0.15 degrees. These simulations are carried on until a complete library is created for all possible error combinations. Lastly, once this library is created, the sample is inspected and compared to the simulations to find the best match. For proof of concept, this is performed using a formula such as

the mean square difference calculation. A diagram of the process is visualized in Figure 1 below.

Inspecting a sample in this fashion solves the ill-posed inverse problem. Since we are matching simulations to experimental results, a maximum amount of information is required for high accuracy. Spectroscopic ellipsometry adds polarization and wavelength information into the simulations while 2-θ scatterometry adds information about the angular spectrum of the diffracted light. Our implementation of FBS adapts standard ellipsometry by creating an incident spatially varying polarization state as opposed to standard ellipsometry's uniformly polarized illumination. In this fashion we can create situations that should null a detector output. If the detector output is completely nulled we do not have direction sensitivity of our inspected process errors when the output deviates from expectation. To establish directional

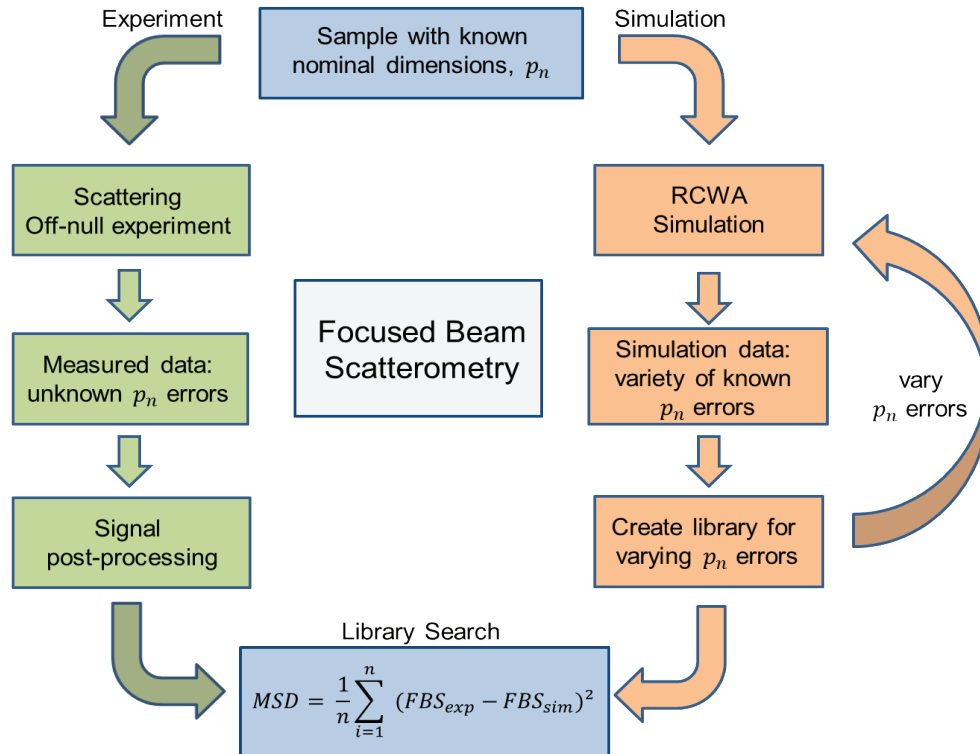


FIGURE 1. The process flow used in Focused Beam Scatterometry

sensitivity we engineer an input polarization state that should yield the output on the CCD in a slightly off-null condition. In this fashion we establish directional sensitivity. Furthermore, off-null inspection theoretically increases the sensitivity of the instrument to individual process parameters [7]. In addition to polarization engineering, we focus the inspection beam through a microscope objective which adds information regarding the angular spectrum just as in 2- θ scatterometry. It has been demonstrated previously that through-focus scanning of the inspected sample enhances the resolution of scatterometry [8,9]. Through-focus scanning has even enabled reliable inspection beyond the 22 nm critical dimension node [10]. In FBS we are investigating precision lateral scanning as another novel approach.

SYSTEM INSTRUMENTATION

The primary instrumentation tasks for this project involve the development of a three-axis closed-loop controlled sample inspection stage and the development of an arbitrary spatially varying polarization state generator. Our sample scanning stage is the Physik Instrumente P-611.3 NanoCube[®] piezo stage [11] which we have placed under closed-loop control using capacitance gauges by Lion Precision [12]. Using a PXI platform by National Instruments with a

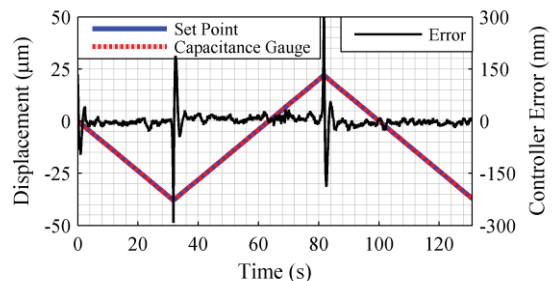


FIGURE 2. Scanning the sample stage under closed-loop control yields low error on the order of the noise floor of the capacitance gauge. A constant phase offset has been removed to evaluate controller error.

high-bandwidth FPGA [13], we can readily achieve sufficiently stable scans in the full 100 μm scanning range. In open-loop, the piezo stage initially displayed the well-known drift and hysteresis effects on the order of hundreds of nanometers. Using a PI-controller, the hysteresis and drift are eliminated in precision scans as shown in Figure 2.

We create a spatially varying polarization state using a phase-only spatial light modulator (SLM). Our approach achieves amplitude control by superimposing a holographic blazed diffraction grating on top of our phase modulation from the SLM [14]. Light is then selectively reflected into the

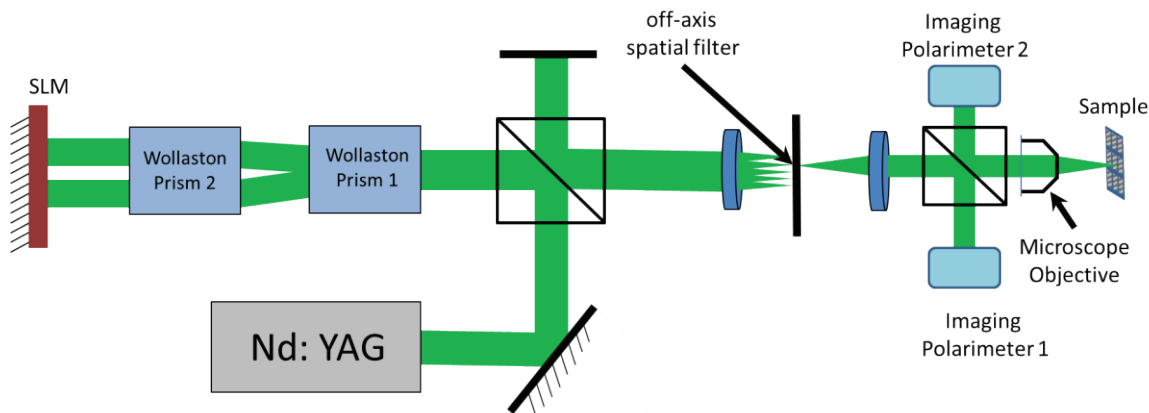


FIGURE 3. The setup used to create arbitrary spatially varying polarization states and measure process parameters. For a more complete description of polarization generation see reference [15].

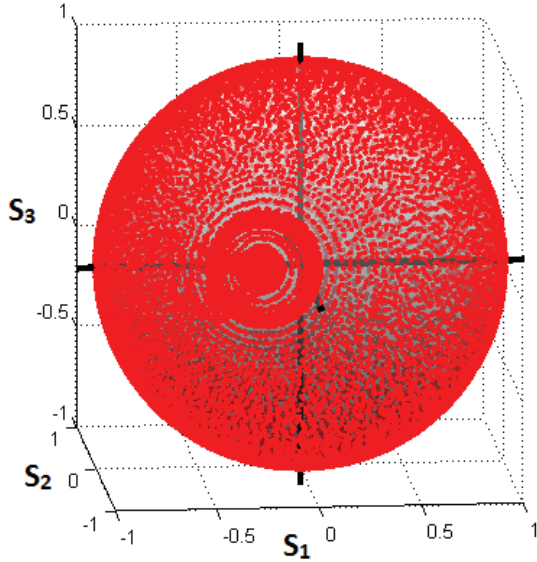


FIGURE 4. The polarization generator has the capability of generating all possible polarization states with greater than 95% degree of polarization.

zero order or diffracted into the first order. Since our encoded signal lies in the first diffraction order, a wide spatial filter in the Fourier plane of the beam filters the signal off-axis. A primary advantage of this implementation is that we separate our encoded signal from pixel diffraction noise from the SLM by using a non-integer number of pixels to represent the blazed grating.

A schematic of the polarization generator is shown in Figure [3]. We split light from a 1064 nm source into its respective TE and TM polarization components using two Wollaston prisms. The first Wollaston separates the beams and the second collimates the beams to be normal with the SLM surface while also ensuring a common path setup. At the SLM, the light is incident on a phase structure affecting two spots. Furthermore, depending on the depth of a blazed grating superimposed on top of the holograms, the amplitude of the light is controlled [14]. After reflecting from the SLM, the light is recombined through the Wollaston prisms into a single beam before a Fourier filter.



FIGURE 5. Arbitrary spatially varying polarizations analyzed using a horizontal linear polarizer (top images) and vertical linear polarizer (bottom images) before being detected by a CCD.

Since the light has been diffracted into the first order to achieve amplitude control, the beam is spatially filtered off-axis in a 4-*f* system. Most of the noise from an encoded hologram will typically lie in the zero order of the Fourier plane. Since the light is off-axis it has been separated from noise which arises from square pixel boundaries. For a more extensive discussion of a similar polarization generator refer to [15].

PRELIMINARY PERFORMANCE

To examine the performance of our arbitrary spatially varying polarization state generator, we first examined its coverage of the Poincaré sphere. This tool allows us to observe all possible polarization states that can be created in one plot. To construct the map we analyze the output light in terms of its Stokes parameters using a polarimeter. The Stokes parameters are defined as

$$S_0 = |E_x|^2 + |E_y|^2 \quad (1)$$

$$S_1 = |E_x|^2 - |E_y|^2 \quad (2)$$

$$S_2 = 2\Re(E_y^* E_x) \quad (3)$$

$$S_3 = 2\Im(E_y^* E_x), \quad (4)$$

where E_x and E_y are the electric field components in a standard Cartesian basis.

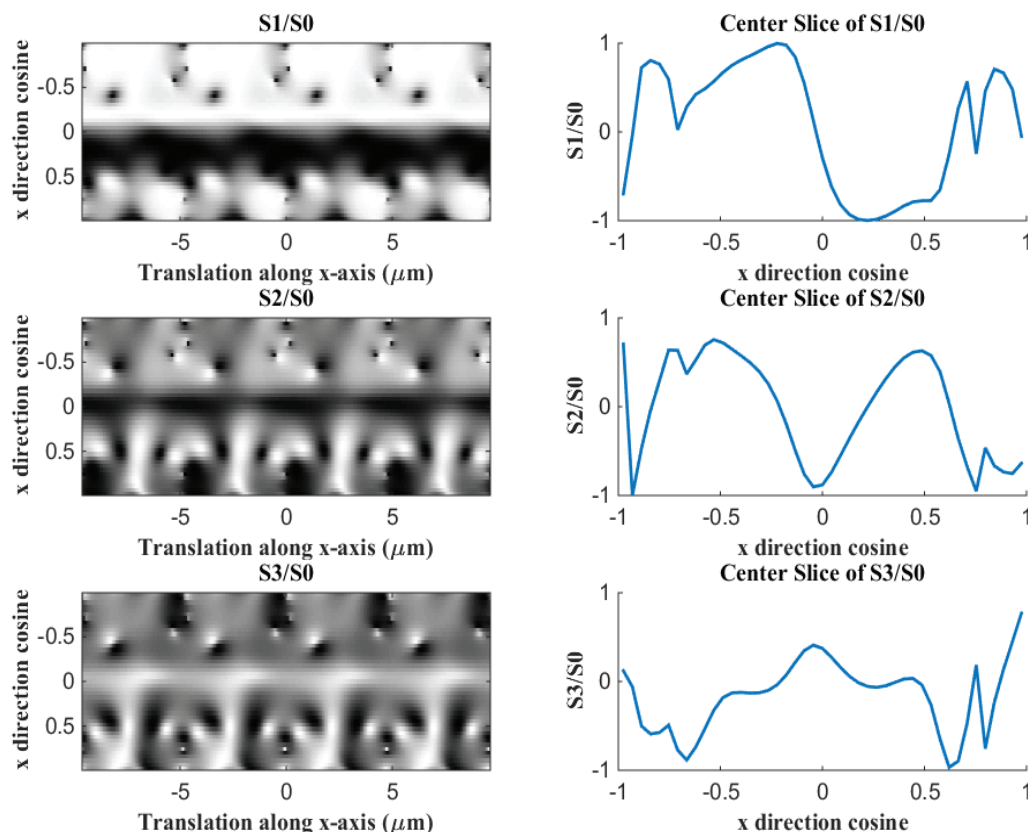


FIGURE 6. The simulated change in normalized Stokes parameters as a function of scanning position. A library containing multiple sets of these three maps will be used to compare against experimental results. (line width = 50 nm, line height = 100 nm, micropitch = 100 nm, macropitch = 4.8 μm , wavelength = 1064 nm, N.A. = 0.9)

The Poincaré sphere represents all linearly polarized components along its equator. Horizontal and vertical polarizations lie along the S_1 direction while $+45^\circ$ and -45° lie along the S_2 direction. Finally, right hand circular and left hand circular lie along the S_3 direction. The complete Poincaré sphere can be seen in Figure 4, demonstrating control of all possible polarization states with greater than 95% degree of polarization on all points. Qualitatively, the performance of the polarization generator is seen in Figure 5. The contrast of the black and white images is created using linear polarizations with high spatial resolution. When analyzed with a horizontal linear polarizer (top images) the inverse image is seen compared to when analyzed with a vertical linear polarizer (bottom images).

Our RCWA simulations have been adapted to allow precision scanning. Figure 6 shows the results of one simulation demonstrating a change in normalized stokes parameters over a 20 μm scanning range. The simulated grating had a 50 nm line width, 100 nm line height, 100 nm micropitch, 4.8 μm macropitch, and the simulated light had a wavelength of 1064 nm with a numerical aperture of 0.9. The polarization of the incident light was a Gaussian TE-component combined with a Hermite-Gaussian TM-component which yields a broad polarization distribution after scattering. In Figure 6, the x-axis designates the lateral scanning position and the y-axis is the cosine of the scattered light in the x-direction. Since the gratings we are inspecting are 2-dimensional, a y-direction cosine would not carry as much information

as the x-direction cosine due to the scattering properties of the sample. After focusing our beam, the final spot on the inspection area is 1 μm . Figure 6 demonstrates a significant increase in the amount of available information for the library match compared to a stationary test. Furthermore, the figure also designates a center slice along the zero scanning position which shows significantly different structure in each normalized stokes parameter.

ACKNOWLEDGMENTS

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REFERENCES

- [1] The international technology roadmap for semiconductors (2013) – *executive summary*, cited March 2015 (<http://www.itrs.net/reports.html>).
- [2] The international technology roadmap for semiconductors (2013) – *metrology summary*, cited March 2015 (<http://www.itrs.net/reports.html>).
- [3] Brown TG, Alonso MA, Vella A, Theisen MJ, Head ST, Gillmer SR, Ellis JD. *Focused beam scatterometry for deep subwavelength metrology*, In: Proc. SPIE Proc. of SPIE Vol. 8949 89490Y; Three-Dimensional and Multidimensional Microscopy: Image Acquisition and Processing XXI, 2014.
- [4] Raymond C. *Overview of scatterometry applications in high volume silicon manufacturing*. AIP Conference Proceedings, 788(394), 2005.
- [5] Peng S, Michael Morris G, *Efficient implementation of rigorous coupled-wave analysis for surface relief gratings*, Journal of the Optical Society of America A; 12, 1087-1096, 1995.
- [6] Lalanne P, Michael Morris G, *Highly improved convergence of the coupled-wave method for TM Polarization*, Journal of the Optical Society of America A; 13, 779-784, 1996.
- [7] Brown TG, Alonso MA, Vella A, Theisen MJ, Head ST. *Weak Measurements applied to process monitoring using focused beam scatterometry*, In: Proc. SPIE Vol. 9050, 90501F; Metrology, Inspection, and Process Control for Microlithography XXVIII, 2014.
- [8] Attota R, Germer TA, Silver RM. *Through-focus scanning-optical-microscope imaging method for nanoscale dimensional analysis*. Optics Letters, 33(17), 2008.
- [9] Attota R, Silver R. *Nanometrology using a through-focus scanning optical microscopy method*. Measurement Science and Technology, 22;024002, 2011.
- [10] Attota R, Bunday B, Vartanian V. *Critical dimension metrology by through-focus scanning optical microscopy beyond the 22nm node*. Applied Physics Letters, 102;222107, 2013.
- [11] URL: <http://www.physikinstrumente.com/product-detail-page/p-61113-201700.html>
- [12] URL: <http://www.lionprecision.com/capacitive-sensors/cpl190.html>
- [13] URL: <http://sine.ni.com/nips/cds/view/p/lang/en/nid/206324>
- [14] Davis JA, Cottrell DM, Campos J, Yzuel MJ, Moreno I. *Encoding amplitude information onto phase-only filters*, Applied Optics; 38(23), 1999.
- [15] Maurer C, Jesacher A, Fürhapter S, Bernet S, Ritsch-Marte M. *Tailoring of arbitrary optical vector beams*, New Journal of Physics; 9, 2007.

Design and calibration of an artifact for evaluating laser scanning articulating arm CMMs used for measuring complex non-concurrent surfaces

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INTRODUCTION

Laser scanners mounted to articulated arm coordinate measuring machines (LS/AACMM) have been recently adopted by the US Army to evaluate ballistic threat mitigation capabilities of human worn body armor. In brief, testing of the armor is performed by placing the article under test against a clay substrate (pre-impact surface), that has been shaped to conform to the concave surface of the body armor, and firing a projectile of a known mass, shape and velocity at it. When the projectile strikes the armor, its kinetic energy is transferred into the armor and clay, resulting in an impact crater (post-impact surface) into the clay known as a back face deformation (BFD), and is an indication of the blunt-force trauma a wearer would experience. The maximum depth of the BFD is used as part of an evaluation criterion to determine if a batch of armor would be placed into service or rejected

In the past, the measurand of the BFD was defined as the distance between the pre-impact surface, and the post-impact surface, at the

point of aim, known as the “Basic Length” FIGURE 1. This measurement was carried out with a measurement device that was similar to a depth gauge called a bridge caliper (BC). With the BC the operator would measure the height of the clay at the point of aim for the projectile, followed by a measurement of the depth at this same point in the impact crater; this method yielded expanded ($k = 2$) measurement uncertainties ranging from 1.6 mm to 1.9 mm, more details can be found in [1, 2]. Among the complications of the BC method is that the BFD at the point of aim—which is a point on the pre-impact surface identified by the laser sight—is not necessarily the maximum BFD depth; see FIGURE 1.

A significant metrological improvement to the BFD measurement process was the introduction of LS/AACMM technology. This allowed the measurand to be unambiguously defined as the longest line segment measured between the pre-impact surface and post-impact surface, known as the “Maximum Distance Length”; see

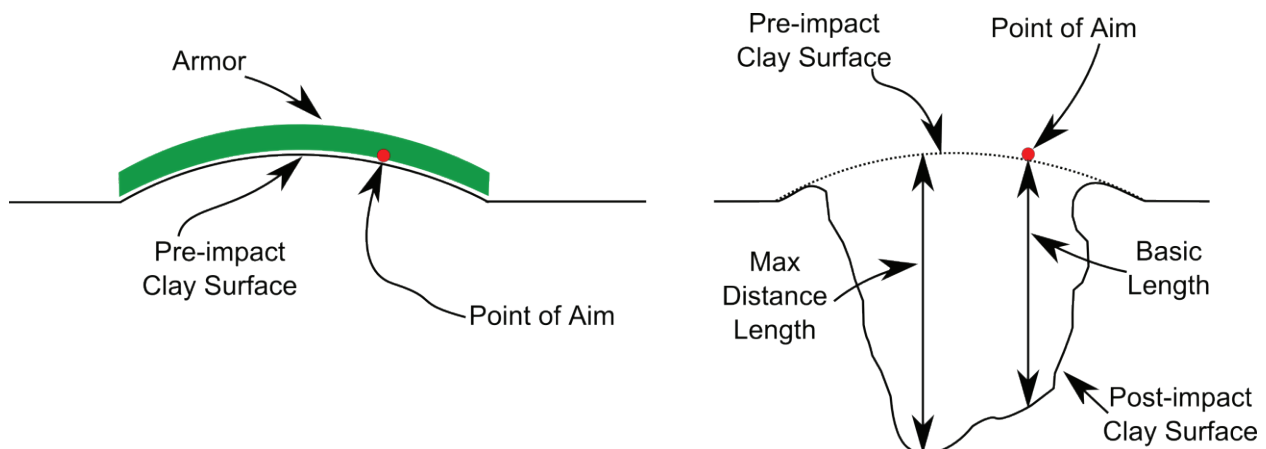


FIGURE 1: (Left) Arrangement of the armor over clay, (Right) Example of impact crater in clay with respect to pre-impact surface

FIGURE 1. Using the LS/ASCMM removed the problematic issue that the point of aim did not necessarily correspond to the point of maximum depth length. Since the location of this maximum value is not known until the post-impact surface is created—and hence the pre-impact surface obliterated—the manual BC method is ineffective in determining the maximum depth length measurand, because detailed topography of the pre-impact surface is required prior to the post-impact surface creation. The LS/AACMM can scan and store detailed information about the pre-impact surface, so it can be recalled and compared against the post-impact surface to calculate the maximum distance length. The initial implementation of the LS/AACMM system significantly reduced the maximum distance length expanded ($k = 2$) uncertainty to 0.37 mm for typical BFD values.

In order for the LS/AACMM system to calculate the BFD value correctly, the pre and post-impact surfaces need to be recorded in a common coordinate system so that the two surfaces can be registered correctly with respect to each other. Three coplanar conical seats on the clay's container provide an interface for the hard probe on the LS/AACMM to establish three discrete points to establish a common datum reference frame (DRF) for both data sets; see FIGURE 3, FIGURE 3, & FIGURE 4.

In brief, the evaluation and measurement of a BFD value is as follows:

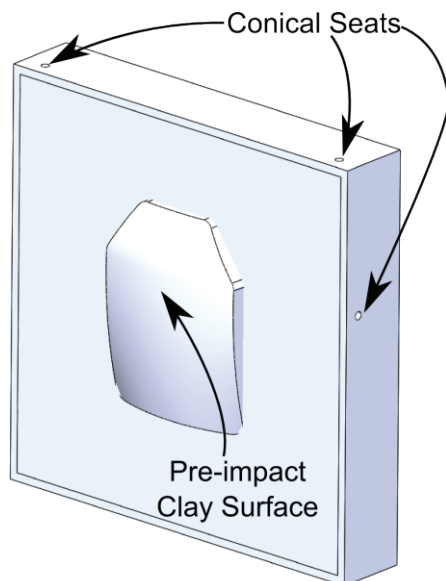


FIGURE 2: Location of conical seats on box relative to pre-impact surface

- 1) A DRF is established by measuring the conical seats on the box using the hard probe of the LS/AACMM.
- 2) The pre-impact surface is scanned and recorded using the LS/AACMM.
- 3) The body armor plate is attached on top of the clay surface.
- 4) A rifle round is fired into the armor and the armor is removed
- 5) The DRF is established again by measuring the conical seats on the box using the hard probe of the LS/AACMM.
- 6) The post-impact surface is measured using the LS/AACMM.
- 7) Mathematical software uses the pre- and post-impact scan data to calculate the maximum BFD distance as defined by Army specifications.

As part of a continuous improvement process, the US Army requested from NIST a measurement check standard to evaluate the performance of the LS/AACMM *in situ* on the live fire test ranges. NIST designed two working prototypes; this paper discusses their designs.

DESIGN OF TEST ARTIFACTS

To adequately test the LS/AACMM used by the Army, the following core design requirements were outlined for the artifacts.

1. Needs to be metrologically traceable
2. A BFD maximum-distance value uncertainty ≤ 0.090 mm (1/4 of initial LS/AACMM measurement uncertainty)
3. Dimensionally stable (less than 0.010 mm per year, for BFD)
4. Contain dimensional features that represent those encountered during measurement
5. Surface features that exercise software for BFD evaluation
6. Similar reflective properties of the backing clay
7. Contain a feature to represent a typical pre-impact test surface
8. Contain a feature(s) to represent a post-impact test surface
9. Contain features used to register pre and post impact scanned data
10. Mimic the BFD measurement work flow.

Considering these design requirements, several design concepts were conceived with two of them developed as prototypes for testing, calibration and delivery to the US Army for

further evaluation, and deployment in their testing facilities.

Design of Concept 1: Kinematic Design

The first design concept consists of two kinematically coupled parts, one representing the pre-impact surface and the other representing the post-impact test surface. The pre-impact surface is modeled after a partial cylindrical section, while the post-impact surface is freeform in nature and modeled after an actual BFD test shot. The test shot chosen by the US Army contains features that display fine structure and sharp changes in gradient that are believed to adequately challenge the LS/AACMM system. These two test surfaces are located and connected together using a kinematic coupling, superimposing one test surface over the other, mimicking how their clay counterparts would be positioned in an actual test (FIGURE 3). The kinematic coupling is a key feature in providing repeatable location of the two parts on the order of $1\ \mu\text{m}$ [3], thus maintaining the calibration between the two parts. This assembly is then mounted into a container that resembles a clay container used in live fire testing. It also contains three coplanar conical seats for the hard probe of the LS/AACMM to establish a datum reference frame (DRF) for each surface. Calibration of this concept would follow the same work flow as measuring a BFD value during live testing by the Army. First, the centers of the three conical seats are measured to establish a DRF. Next, the part representing the pre-impact surface is scanned. Then the pre-impact surface is removed, exposing the post-impact surface. Finally, the conical seats are measured again,

and the post-impact surface is scanned.

Design of Concept 2: Dual Chamber Design

The second design concept is functionally similar to the previous one, but rather than have the pre-impact surface physically superimposed over the post-impact surface, the two surfaces are contained in their own chambers with their own DRFs. Since the DRF has to be established before a scan is performed, each surface can be scanned independently as long as the DRF associated with that surface is used. The software will automatically superimpose the pre-impact surface over the post-impact surface to calculate the BFD.

The surfaces for both of these design concepts have been media blasted using 400 grit aluminum oxide powder to provide a surface finish that is cooperative with the LS/AACMM system, and similar to the clay surface.

CALIBRATION OF TEST ARTIFACTS

Calibration of these artifacts was performed on a high accuracy Leitz PMM-C 8.10.6 coordinate measurement machine with a tactile touch probe. A stylus with a 0.5 mm diameter tip was used to probe the fine structure of the post-impact surface, which was milled using a 0.0625 inch diameter ball nose end mill. To adequately digitize the surfaces a point measurement density of 20 points per mm was used. However the area which the BFD could potentially be located in spans $30 \times 30\ \text{mm}$. Measuring this area using a high point density would be time consuming and impractical. The solution was to use a course measurement to identify a few candidate locations that could possibly contain

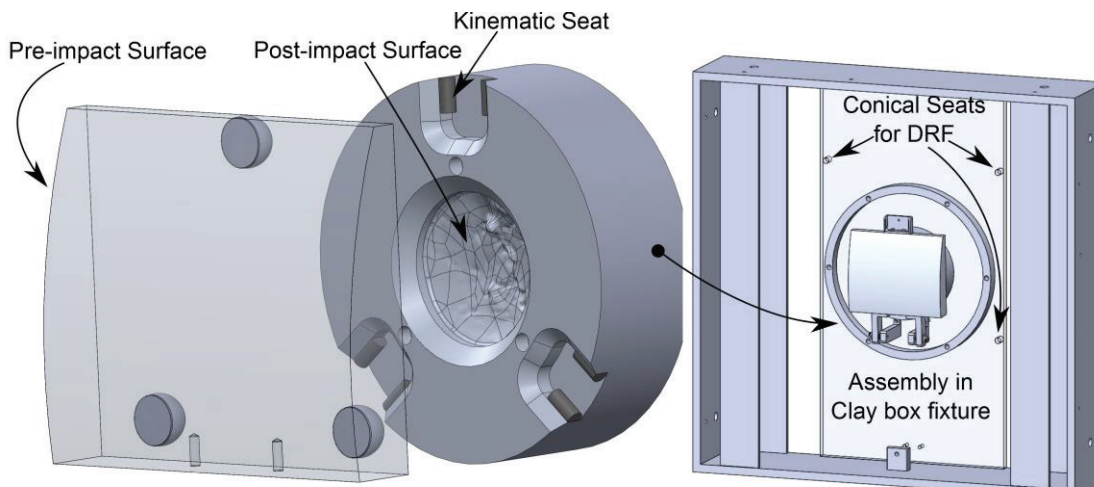


FIGURE 3: (Left) Detailed view of Kinematic design, (Right) Position of artifact in box fixture

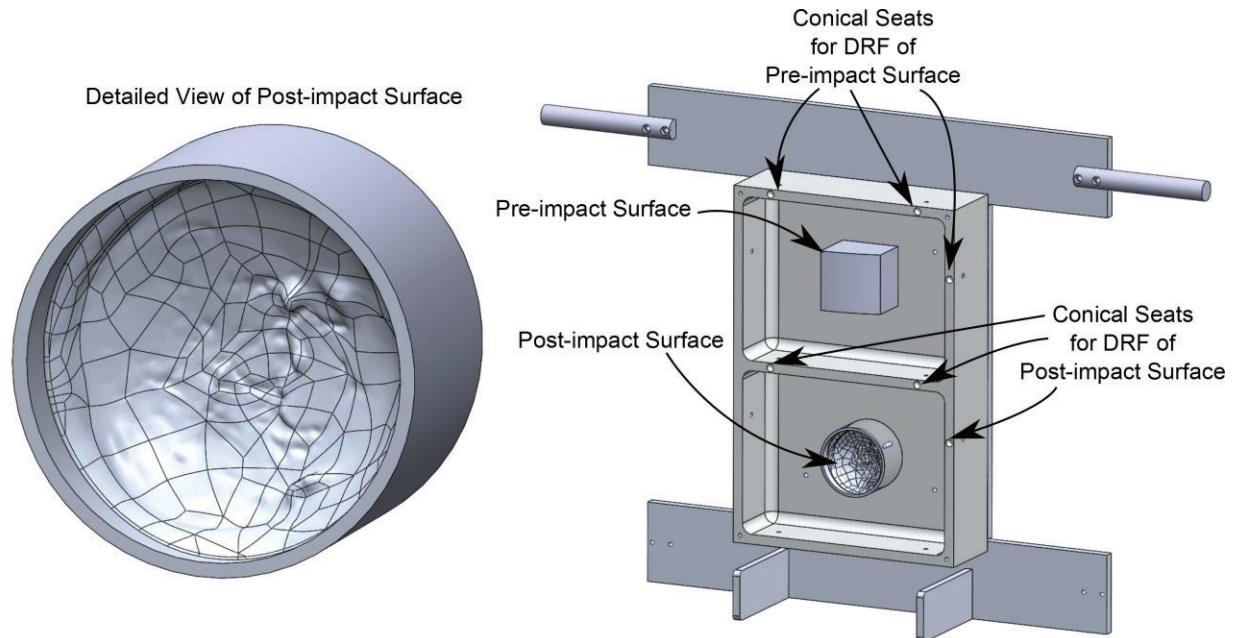


FIGURE 4: (Left) Detailed view of Post-impact surface for Dual Chamber design, (Right) Assembly of pre and post-impact surfaces in dual chamber fixture with respect to DRFs

the largest BFD value and then measure those few locations with a high point density. These artifacts were placed in the CMM such that the surfaces being measured were approximately perpendicular to the Z axis of the machine.

The data collected was post-processed by NIST's own mathematical software algorithms, and not the proprietary software used by the Army, allowing an independent check even of the software processing/smoothing algorithms used during actual BFD testing.

CALIBRATION UNCERTAINTY and RESULTS

The data captured by the CMM was post processed using an NIST algorithm designed to yield the value of the maximum distance length measurand [1]. The calibrated BFD values for the two artifacts described in this paper are outlined in *Table 1*. The NIST measurement uncertainty budget for the kinematic design (KD) and the dual chamber (DC) design is shown in *Table 2*. The expanded uncertainties are 18 % and 27 % of the initial design target uncertainty of 90 μm , respectively.

Table 1: Calibration results for kinematic design (KD) and dual chamber (DC) BFD artifacts (mm)

	BFD Value	Uncertainty ($k=2$)
Design 1 (KD)	47.704	0.016
Design 2 (DC)	41.240	0.024

Table 2: NIST measurement uncertainty budget

	KD std. unc. (μm)	DC std. unc. (μm)
Local CMM repeatability on pre- & post- impact surfaces	0.3	0.6
Projection of coordinates from stylus center to BFD surface	4.9	4.9
Z-axis systematic errors from calibrated step standards	0.1	0.1
Coordinate system and kinematic reproducibility	4.2	6
Thermal uncertainties:		
Due to Uncertainty in Temperature	< 0.1	< 0.1
Due to Uncertainty in CTE	< 0.1	< 0.1
BFD algorithm accuracy	< 0.1	< 0.1
Loading deformation on BFD	5.1	9.0
Combined standard uncertainty	8.2	12
Expanded uncertainty ($k = 2$)	16	24

ARMY TESTING RESULTS

Initial testing results conducted at the Army's Aberdeen Test Center (ATC) are summarized in *Table 3*. The measurements consist of 48 BFD values (from eight different technicians) performed both in ATC's laboratory, using equipment and procedures similar to the live-fire test ranges, and on the actual live-fire test ranges. Examination of the estimated errors (each error being the ATC measured value minus the NIST calibrated value) provides insight into the effectiveness of the BFD artifacts

and an initial view of ATC's current measurement capability. Table 3 shows a summary of the results, presented as (1) the 95th percentile of the error distribution and (2) twice the root-mean-square of the estimated errors. The RMS value was computed, as opposed to the standard deviation, because the RMS evaluates deviations from the calibrated value including systematic errors, as opposed to just the deviations from the mean error, which are evaluated in the standard deviation. A complete ATC uncertainty budget would require more measurements spanning a wider range of influence quantities and also combining the NIST calibration uncertainty. Nonetheless the currently available ATC results show excellent agreement with the NIST calibrated values, especially notable in consideration of the high-volume measurement environment of the actual test ranges. The significant improvement in ATC's measurement capability reflects an ongoing effort for continuous metrological improvement of their BFD dimensional measurement capability.

Table 3: Initial ATC test results: 2 RMS and 95th percentile of (ATC value – NIST value) (mm)*

	KD 2*RMS	KD 95 th %	DC 2 *RMS	DC 95 th %
ATC Lab	0.040	0.047	0.037	0.039
ATC Live Fire Range	0.063	0.060	0.109	0.094

CONCLUSION

Two artifacts satisfying all of the core design requirements have been designed, developed, calibrated, and delivered to the US Army. Feedback from the Army noted that both designs performed well with their LS/AACMM systems. The kinematic design provided more repeatable results when measured in ATC's live fire range, when compared to the dual chamber concept. One likely reason is that the pre and post-impact surfaces share a common DRF. The other is the short metrological loop that is maintained by the kinematic coupling of the pre and post-impact surfaces. However the kinematic design was very sensitive to incomplete seating of the kinematic coupling. If the not properly seated, the kinematic design would produce an incorrect BFD value, something that the dual chamber design wasn't subjected to since it has no moving parts. . Comparison with an initial set of Army BFD measurements shows excellent agreement with NIST results and significant metrological

improvements relative to their initial implementation of the LS/AACMM system.

DISCALIMER

Certain commercial equipment, instruments, or materials are identified in this paper in order to specify the test and measurement procedure adequately. Such identification is not intended to imply recommendation or endorsement by the National Institute of Standards and Technology, nor is it intended to imply that the materials or equipment identified are necessarily the best available for the purpose.

REFERENCES

- [1] K. Rice, M. Riley, A. Forster, S. D. Phillips, C. M. Shakarji, D. Sawyer, *et al.*, "Dimensional Metrology Issues of Army Body Armor Testing," National Institute of Standards and Technology, Gaithersburg, MD 2010 (unpublished).
- [2] "Department of Defense Test Method Standard For Performance Requirements and Testing of Body Armor," in *MIL-STD-3027*, ed: Department of Defense, 2008.
- [3] A. Slocum, *Precision Machine Design*. Dearborn, Michigan: Society of Manufacturing Engineers, 1992.

DESIGN OF A NANOMETER-ACCURATE FORM MEASUREMENT MACHINE FOR SEGMENTED LARGE TELESCOPE MIRRORS

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EXTREMELY LARGE TELESCOPES

The planned extremely large telescope observatories with segmented giant primary mirrors will require metrology with nanometer accuracy. The two extremely large ground telescopes, the European Extremely Large Telescope (E-ELT [2]) and Thirty Meter Telescope (TMT [3]), feature telescopes with 39 meter and 30 meter primary mirrors (M1), respectively. The E-ELT consists of 798 hexagonal segments, while the TMT consists of 492 hexagonal segments. The segments of both telescopes are approximately 1.5 m from corner to corner.

This paper describes the preliminary design of a measurement machine for these segments. A full length, detailed description of the design can be found in Bos et al. [1].

Metrology requirements

Metrology equipment with nanometer-level uncertainty is required for the final metrology of the primary mirror segments, which is challenging in view of the segment dimensions. Based on a TNO assessment, the metrology uncertainty of the form error of the segments (including focus, astigmatism and trefoil) should not exceed 15 nm RMS. Non-contact metrology is required to prevent damaging the optical surface of the segments.

Current metrology methods

Many techniques have been developed to measure mirrors and mirror segments for the existing ground-based telescopes. One of the most promising techniques is measuring the mirror segments using an imaging technique: Fizeau Interferometry with Computer-Generated Hologram (CGH) correction [4]. Although measurement times are very short, setup, alignment, and calibration times will be long and thereby increasing cost of operation. Moreover, the need for a few dozen different surface-specific CGHs makes this method non-universal.

Single-point scanning technique

As an alternative to the current metrology methods, a measurement machine is proposed based on the principle used in the NANOMEFOS [5, 6]. However, the rotating segment concept of the NANOMEFOS was rejected because the disturbances acting on the segment during rotation of the segment are not representative compared to its future use in the telescopes, and also increase the measurement uncertainty. Therefore, the chosen concept is a more classical orthogonal setup with a translating probe and a stationary segment. This results in a more complex 3D metrology problem. By adding a separate metrology system, this measurement machine, based on a single-point scanning technique, will be capable of measuring large measurement volumes within the required uncertainty. Overall measurement time (including handling of the segment assembly) and cost per measurement are minimized compared to other metrology solutions as alignment effort is reduced, the measurement is fully automated, and no dedicated artifacts for each segment are required. This concept has the following advantages over other principles.

- The typical measurement time of one segment is 15 minutes, using a 3 mm lateral resolution.
- After initial calibration, the operational cost of the machine are low, as no dedicated artifacts or CGHs are required in-between measuring consecutive segments.
- Being able to measure all kinds of reference artifacts, the machine has the potential to be traceable to the SI units of length.
- The implementation of a tactile precision probe eliminates the need for the CMM in the earlier manufacturing stages.

CONCEPTUAL DESIGN

In the proposed machine design, a non-contact optical probe is translated in x and y (Fig. 1). By applying a non-contact optical probe with a dynamical range of at least 5 mm and an acceptance angle of at least $\pm 1^\circ$ no vertical translation (z) or probe rotations (tip, tilt) are necessary, as the maximal sag and slope of a (TMT) segment is approximately 4.2 mm and $\pm 0.7^\circ$ respectively. This minimizes the number of motion axes.

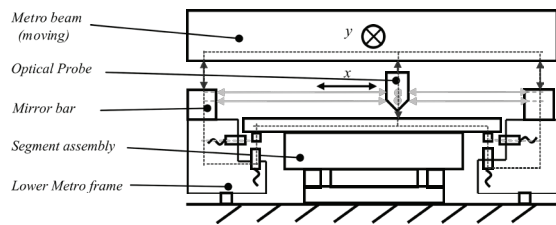


FIGURE 1. Machine concept: a non-contact optical probe is translated in a plane, using an x - and y -translation.

All degrees of freedom of the segment assembly are measured relative to a lower metrology frame, using capacitive probes measuring on metal blocks bonded to the underside of the segment. The z -direction of the non-contact optical probe is measured via a metrology beam to the lower metrology frame using three interferometer axes. Reflective bars are bonded to the base block of the lower metrology frame. The x , y , tip, and tilt movements of the non-contact optical probe are measured twice relative to these reflective bars using eight interferometer axes, to obtain redundant data and thus reduce the measurement noise (see the 3D metrology setup depicted in Fig. 2).

The basic difference between a standard CMM and this measurement machine is the measurement of all degrees of freedom of the segment relative to a metrology frame, therefore shortening the metrology loop and separating it from the structural loop, reducing the uncertainty. Moreover, the x , y , z , ϕ and ψ -coordinate of the optical non-contact probe are measured directly on the probe, relative to the lower metrology frame.

The machine design is shown in Fig. 3. It measures 3.0 m wide, 2.6 m deep and is 1.5 m high. The total mass is approximately 12.5 tons.

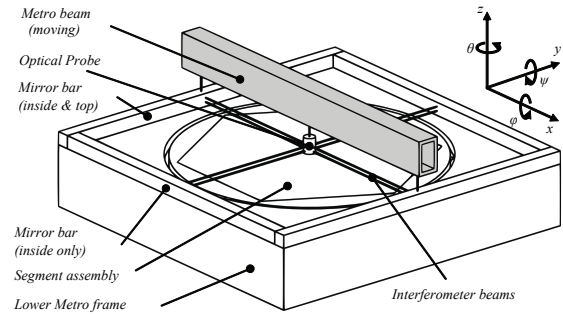


FIGURE 2. 3D metrology concept. All degrees of freedom of the segment assembly are measured relative to the lower metrology frame.

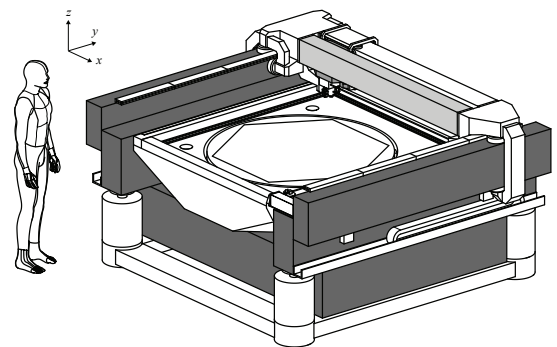


FIGURE 3. Machine design.

The design consists of two main systems: the motion system and the metrology system. The motion system mainly consists of an x -stage and a y -stage. The metrology system mainly consists of a lower metrology frame, the metrology beam (which is connected to the y -stage), the interferometry system, and an optical and tactile probe. Furthermore, to make fast and repeatable mounting of the segment assembly possible, a segment assembly mounting tool is designed in combination with an extractor mechanism. The mounting tool can be used throughout the manufacturing process of the segment, once its support structure is integrated. The mounting tool is mounted in a statically determined way on the machine base using three balls and v-grooves. Not three, but six v-grooves are mounted on the base at 60° each, so it is possible to rotate the segment in steps of 60° , allowing calibration techniques such as the multi-step method.

The motion system and the metrology system are both connected to the base, which consists of an assembly of granite blocks, which is supported by four passive vibration isolators.

MOTION SYSTEM

The motion system positions the optical probe relative to the segment in two degrees of freedom. The optical probe is positioned in the measurement plane (x and y) via an x -stage and a y -stage. The x -stage is guided via air-bearings on an Alumina (Al_2O_3) guidance beam. The guidance beam is connected to two y -carriages guided by air-bearings that together form the y -stage. To minimise distortions and hysteresis, the stages have force-closed air-bearing setups with separate position and pre-load frames. Ironless linear motors are applied and high-resolution linear encoders are used for position feedback.

x -stage

The x -stage (Fig. 4) provides the optical probe with six degrees of freedom. It constrains these such that sub- μm uncertainty is achieved at the probe tip. The x , y , z , ϕ and ψ -direction are measured by the interferometry system on the optical probe. The x -stage is aligned with four air-bearings to a vertical plane of the guidance beam, constraining it in the y , ϕ and θ -direction, but, over-constraining it. The position frame to which the bearings are connected has an internal degree of freedom to eliminate this over-constrained direction. Two bearings aligned with the lower horizontal plane constrain the x -stage in z and ψ -direction. The x -direction, which is the direction of motion, is constrained by the linear motor with its servo-stiffness. The mass of the x -stage is approximately 15 kg and it is designed to move with accelerations up to 10 m/s^2 .

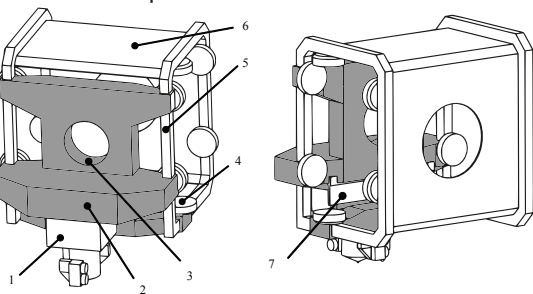


FIGURE 4. x -stage assembly. 1: optical probe, 2: position frame, 3: weight reduction, 4: air-bearings, 5 & 6: pre-load frame, 7: linear motor.

y -stage

The y -stage moves the probe in the direction perpendicular to the x -stage with a low velocity. The guidance beam of the x -stage and the Sintered Silicon Carbide (SSiC) metrology beam form the

y -stage, together with two carriages. In Fig. 5, both the front view (top) and the rear view (bottom) are given. The mass of the y -stage is approximately 350 kg.

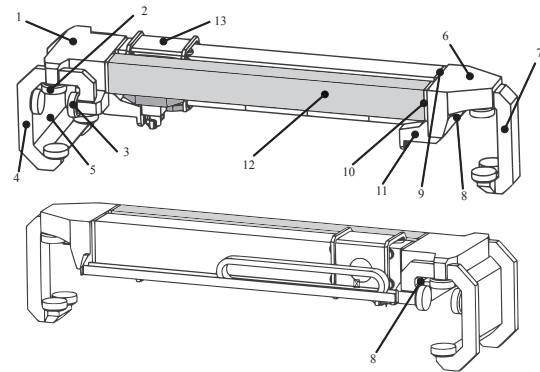


FIGURE 5. y -stage assembly. 1: position frame, 2 & 3: air-bearings, 4 & 5: pre-load frame, 6: position frame, 7: pre-load frame, 8: linear motors, 9: guidance beam support interface, 10: metrology beam support interface, 11: linear encoder read-head, 12: SSiC metrology beam, 13: x -stage.

METROLOGY SYSTEM

The metrology system measures the position of the optical probe relative to the segment in all the degrees of freedom using a 12-axis interferometer system, a metrology beam moving in the y -direction and a stable lower metrology frame. An exploded view of the metrology system is depicted in Fig. 6. SSiC is the material of choice for the metrology beam, due to its excellent thermal and mechanical properties. The lower metrology frame is made of Zerodur, due to its near zero coefficient of thermal expansion. Nanometer-level stabilities are expected.

Lower metrology frame

The lower metrology frame is depicted in Fig. 7. The lower metrology frame consists of a block of Zerodur with a hole and chamfered lower corners and part of the edges (1). To this block, four Zerodur mirrors are bonded with an adhesive. The two bars that act as a reference for the y -direction interferometer measurement (2) need to be straight and polished to optical quality on the inside. The two bars that are both a reference for the x - and z -direction (3) need to have a straight upper side and inside polished to optical quality. The two Zerodur precision scales (4) that act as a position feedback for the y -stage linear motors are bonded on top of the two Zero-

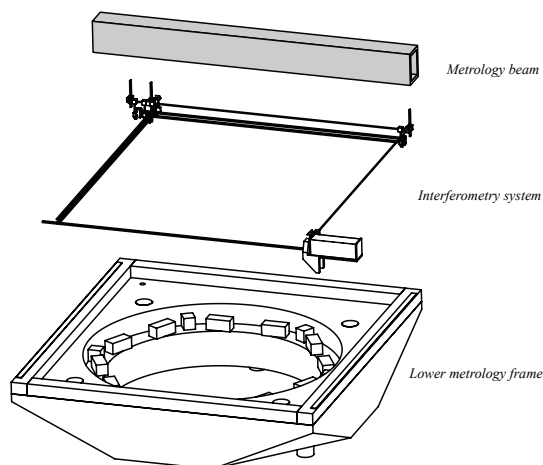


FIGURE 6. Exploded view of the metrology system.

dur bars by fusion bonding. Six capacitive probes with (automated) adjustment systems (5) have to be mounted on the Zerodur block. As an addition, twelve (adjustable) edge sensors (6) can be mounted on the Zerodur frame to obtain information about the real mirror surface positions, as if the segment was positioned in the telescope. Together with the four zero-plane reference mirrors (7), the position and movement of the segment with respect to the metrology frame is known. A sphere (8) is bonded to the Zerodur block to act as an x , y , z nulling-target for the non-contact probe in the home-position. The lower metrology frame is mounted on the granite base block at three points, using three hinged blocks (9). The mass of the lower metrology frame assembly is 1950 kg and the first resonance is a torsional mode at 110 Hz. The second resonance is a torsional mode at 160 Hz.

Interferometry system

The interferometry system, depicted in Fig. 8, measures the displacement of the optical probe relative to the reference mirror bars on the lower metrology frame. The x , y , ϕ , and ψ are measured directly, but, depending on the probe position, with a relatively large distance through air. The z displacement of the optical probe is measured indirectly via a metrology beam to the mirror beams on the lower metrology frame. However, the distance through air is shorter. The x , y , ϕ and ψ coordinates are measured twice to get a weighted distance measurement and to obtain redundant data, lowering the measurement noise. This results in eight axes for the interferometer

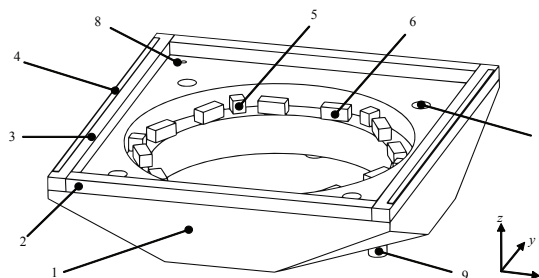


FIGURE 7. Lower metrology frame assembly. 1: Zerodur block, 2 & 3: mirror bars, 4: linear encoder precision scales, 5: capacitive probes, 6: edge sensors, 7: zero-plane reference mirrors, 8: nulling-target, 9: hinged support blocks.

system measuring displacements and tilts. The z -measurement of the optical probe to the metrology beam is the ninth axis and adding the two z -measurements at the ends of the metrology beam to the mirror bars of the lower metrology frame completes the metrology. This results in the need of an eleven-axis interferometry system. Nowadays, twelve-axis interferometer systems (2 x 6) are commercially available using only one laser. The twelfth interferometer axis can be used to measure the relative air-index variations by measuring from one side of the inner-surface of the Zerodur mirror bars to the other. A heterodyne, double-pass, plane interferometer system is used in the design of this machine.

Probes

A non-contact probe based on the differential confocal principle [7] is well-suited for the measurement machine. This probe has a 0.1 nm resolution, an uncertainty of ≤ 10 nm RMS for medium freeform surfaces ($< 2^\circ$), 5 mm measurement range, $\pm 5^\circ$ acceptance angle with tilt dependency compensation, and 1.6 mm distance between lens and segment. The measurement range of 5 mm can be increased to a safer 8 - 10 mm, while the acceptance angle of $\pm 5^\circ$ could be reduced.

A tactile precision probe is added to the design, mounted next to the optical probe, to be able to measure a ground segment surface with an accuracy of $\leq 0.1 - 0.2 \mu\text{m}$ RMS. The interferometer system can still be used to obtain information about the x , y , ϕ , and ψ movement but with a small offset, only resulting in a small error. The tactile probe requires an accurate z -guidance with a stroke of about 15 mm.

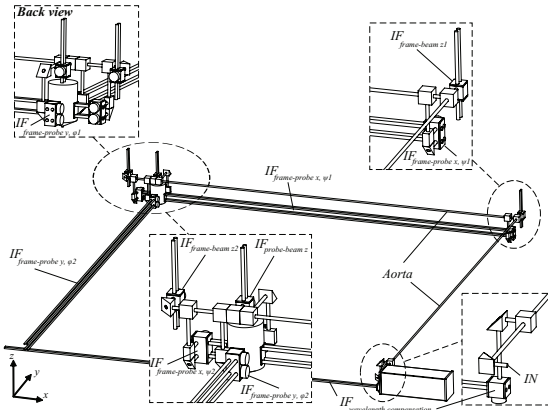


FIGURE 8. The 12-axis interferometer design. The interferometer system is divided into two levels: the upper level with the laser source, the optical supply line for all the interferometers (the Aorta) and the z -interferometers. The lower level consists of four (double-)interferometers for the x , y , ϕ and ψ measurement and the twelfth axis for measuring air-index variations.

MEASUREMENT UNCERTAINTY

The measurement uncertainty of the machine after calibration is estimated to be 11.2 nm RMS. This value is based on a first order uncertainty estimation, experience [5, 6, 7], knowledge of the designed system, estimates about the accuracies of the principles used, and estimates about the stability of the metrology loop [1]. It is expected that the individual error sources can be measured using well-known artifacts such as straight-edges, optical flats and true-squares, using self-calibration principles [8]. Also well-characterized (spherical) mirrors can be used as end-to-end calibration artifacts, together with stitching to cover the entire measurement volume with smaller mirrors. A 1.4 m spherical reference mirror, calibrated in an interferometric test tower [4] with similar radius of curvature could be used as 'black box' calibration method. Calibrating the artifacts to the required uncertainty level is of course challenging, as reference standards with the required uncertainty over these sizes are not trivial. More research needs to be done on how to calibrate the machine, but the machine has the potential to be traceable to the SI unit of length.

In the design of the machine, redundant information in several directions is available, for instance for the degrees of freedom of the segment by applying twelve capacitive probes, where only six

are required as a minimum. This redundant data can be used for averaging, decreasing the measurement noise and increasing the machine's repeatability. Being able to turn the segment assembly in multiple steps of 60° can decrease the uncertainty, aiding to distinguish several Zernike orders and possible causes of segment deformation due to misalignment and mounting. The measurement strategy could be to measure a coarse grid over the machine before starting a high-resolution measurement, to be able to distinguish thermal drift from segment form.

REFERENCES

- [1] Bos, A., Henselmans, R., Rosielle, P., and Steinbuch, M., Nanometre-accurate form measurement machine for E-ELT M1 segments, *Journal of Precision Engineering*, 2015; 40: 14-25
- [2] European Extremely Large Telescope (E-ELT), www.eso.org/public/usa/teles-instr/e-elt/
- [3] Thirty Meter Telescope (TMT), www.tmt.org
- [4] Burge J.H., Zhao, C., Dubin, M., Measurement of aspheric mirror segments using Fizeau interferometry with CGH correction, *Proc. of SPIE 7739, Modern Technologies in Space- and Ground-based Telescopes and Instrumentation*, 2010; 773902: 1-15
- [5] Henselmans, R., Cacace, L., Kramer, G., Rosielle, P., and Steinbuch, M., The NANOMEFOS non-contact measurement machine for freeform optics, *Journal of Precision Engineering*, 2011; 35: 607-624
- [6] Henselmans, R., Non-contact measurement machine for freeform optics, PhD Thesis, Eindhoven University of Technology, 2009.
- [7] Cacace, L.A., An Optical Distance Sensor: Tilt robust differential confocal measurement with mm range and nm uncertainty, PhD Thesis, Eindhoven University of Technology, 2009
- [8] Evans, J., Hocken, R., and Estler, W., Self-calibration: Reversal, Redundancy, Error Separation, and 'Absolute Testing', *Annals of the CIRP*, 1996; 45/2: 617-634

TOPOLOGY OPTIMIZATION FOR THE CONCEPTUAL DESIGN OF PRECISION MECHATRONIC SYSTEMS

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INTRODUCTION

The design of high-precision (nanometer to sub-nanometer) motion systems increasingly relies on a multidisciplinary design approach from the outset [1]. This is because conflicting requirements necessitate a balanced trade-off between, e.g., thermal, mechanical and control specifications, which is hard to reach sequentially. The advent of additive manufacturing techniques that allow unprecedented design freedom, has a significant impact on the precision industry. It is expected that within years this freeform manufacturing technique will allow the production of sophisticated device components with integrated functionality, which meet the demands of precision technology. Topology optimization methods [2], which deal with the problem of optimizing the material distribution in a component, are a very natural counterpart to the freeform nature of additive manufacturing.

Topology optimization is already established as a design tool in the aerospace and automotive industries where it is mostly used to optimise components for minimal mass subject to stiffness or limit stress requirements and to generate initial (one might say conceptual) shape designs. On the other hand, significant developments are required to bring these techniques to a level where industry can apply them to complex multidisciplinary design problems with more sophisticated performance metrics, such as closed-loop control performance.

In this paper we specifically consider the topology design of components involved in high-precision and high-speed positioning tasks, such as positioning stages. This means there is a strong link between the dynamics of the component and the achievable closed-loop performance (such as bandwidth) and parts cannot be modelled as rigid bodies. From a control systems point of view

it is often advantageous to design parts with high eigenfrequencies. This criterion by itself, however, does not convey any information about mode *shapes* and their effects on the frequency response functions as seen by the controller, and may therefore lead to conservative or poorly balanced designs. For this reason, a topology optimization formulation is developed which directly designs for closed-loop performance (based on first ideas discussed in [3]). In our formulation the component should simultaneously satisfy other stiffness and mechanical stability requirements with respect to the deformations that occur during motions and accelerations. One reason is that positioning accuracy suffers when the relative displacement between a point of interest and a position sensor becomes too large.

For reasons of space we only briefly discuss the modelling approach, to then move on to the results and a brief discussion of the first findings.

SETUP AND MODELLING

Throughout this article we consider the simplified 2D model of a positioning stage as shown in Fig. 1. This model represents a positioning component in a precision device. The stage is treated as a free-floating body, positioned by actuator forces F_x and F_y . The forces F_x always act simultaneously, whereas $F_{y,1}$ and $F_{y,2}$ can be applied separately, leading to three independent force inputs. Furthermore, there is a contactless measurement setup. Here, it is assumed that the coordinate x is measured as the average between the two sensors, whereas the coordinates y_1 and y_2 are measured independently. Thus, there are also three independent measurements. Note that depending on the position of the stage with respect to the metrology frame, the sensors see different points on the structure, giving rise to position-dependent dynamics (see page 4).

Together, the inputs and outputs allow position-

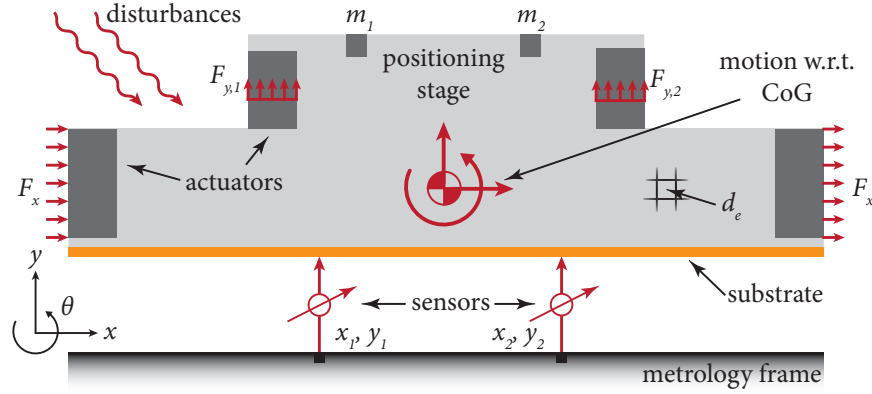


FIGURE 1. Simplified representation of a free-floating motion stage which can be controlled in three planar degrees of freedom (DoFs). Each of the two sensors measures x and y displacements. Two actuators each are available in the x and y directions to provide 3-DoF controllability. The light grey area may be modified by topology optimization.

ing of the stage in any of the three rigid-body degrees of freedom (defined as x , y and θ) and allow uniquely measuring this position. Each of the actuators is defined by a mass associated with the dark gray areas in the diagram. Finally, there is a substrate layer which may have different material properties.

Mechanical modelling

The mechanical system is modelled using a finite element (FE) discretization. In this 2D case, bilinear quadrilaterals are used. It is assumed that the actuator forces are uniformly distributed throughout the actuator bodies. The dynamic behavior of the stage is described by a reduced-order model which is based on the first 10 flexible modes of the stage¹ in addition to the 3 rigid-body (RB) modes, using a mode truncation approach. This frequency-domain model will be called $P(j\omega)$ and describes the frequency responses from the actuators to the sensors.

Static decoupling of the three RB motions is obtained by computing pre and post compensation matrices W_A and W_S [4] to result in a statically decoupled plant $G(j\omega)$:

$$G(j\omega) = W_S P(j\omega) W_A.$$

This means the compensated system has new sets of input and output signals, for which it holds that a constant input to each of these gives a

¹Based on the fact that the 10th natural frequency is about 15 times higher than the expected controller bandwidths.

constant RB acceleration, which affects only the corresponding RB output. The advantage is that this allows a sequential loop closure approach using independent single-input-single-output controllers.

Control design

Each rigid-body motion is controlled by a PID controller of simple structure, according to:

$$C_i(s; k, \omega_b) = k \frac{s + \frac{1}{5}\omega_b}{s} \frac{3s + \omega_b}{s + 3\omega_b} \frac{5\omega_b}{s + 5\omega_b},$$

where ω_b is the controller bandwidth and k is the controller gain. This controller structure is often applied in the field of motion systems [1] (e.g., $\frac{1}{ms^2}$ type systems) and provides integral action up to $\frac{1}{5}\omega_b$, phase lead between $\frac{1}{3}\omega_b$ and $3\omega_b$ and first order roll-off beyond $5\omega_b$. This structure leads to sufficient phase lead in the crossover region, low frequency disturbance rejection and limited high frequency control action.

For each of the control loops a controller is designed which minimises the closed-loop sensitivity transfer function at 10 Hz, while restricting peaks in the sensitivity function to a maximum of 6 dB.

Topology parameterization

The structure is parameterized using the density approach in topology optimization [2]. This implies that each finite element is assigned a design value d_e between 0 and 1 (see Fig. 1). Then, the properties of that finite element are scaled

with the design value according to an interpolation function. We use a modified form of the SIMP (Solid Isotropic Material with Penalization) approach which is widespread in topology optimization [2]. Specifically, for an element e , this results in:

$$K_e = d_e^3 K_e^0, \quad (1)$$

$$M_e = \begin{cases} d_e M_e^0, & d_e > 0.1, \\ d_e^5 M_e^0, & d_e \leq 0.1, \end{cases} \quad (2)$$

in which K_e^0 and M_e^0 are the element matrices for solid material. These interpolations ensure that intermediate-density elements tend to underperform (i.e., due to a poor stiffness-to-mass ratio), encouraging a black-and-white design. The special mass interpolation (2) ensures that for low design values the element mass tends to zero faster than its stiffness. This is necessary to suppress localized vibration modes in low-density areas [5].

Referring to the diagram (Fig. 1), only the light gray areas have been parameterized, which implies that only the topology in these areas may be modified. Square elements were used, giving 116 elements along the x -dimension and 34 elements along the y -dimension.

Furthermore, the actuator masses are defined as a fraction of the overall structural mass, since it is known that actuator sizing depends on the mass of the stage. Hence, a lighter structure requires lighter actuators and this dependence is accounted for.

Design sensitivity analysis

The optimization procedure used to arrive at improved designs relies heavily on design sensitivity analysis. For a given design, say the initial topology with initial controller parameters, the performance and all constraint functions are computed. Then, the gradients of these functions with respect to all design variables are computed, these variables being the topology design variables d_e , $e = 1, \dots, n_d$ and the controller variables $k_i, \omega_{b,i}$, $i = 1, 2, 3$.

Although finite-differencing is a simple technique to obtain sensitivity information, it is prohibitively expensive for many design variables with functions that depend on expensive finite element computations. Therefore, analytical sensitivities are computed. Due to page restrictions it is not possible to describe these sensitivity computa-

tions in detail. Instead, we give a sketch of the procedure:

For the functions related to closed-loop system dynamics:

- The sensitivity of each mode shape with respect to a design variable d_e can be computed using *adjoint sensitivity analysis*;
- These *mode shape sensitivities* are used to evaluate the sensitivity of the transfer function $P(j\omega)$ as well as the pre and post compensation matrices W_A and W_S . This information, combined with the sensitivity of the controller $C(s)$ to parameter changes, is used to compute the sensitivity of functions related to the loop gain.

For the functions related to mechanical performance (static FE computations):

- The sensitivity of a structural deformation for a given load case can be computed using an adjoint sensitivity analysis;
- This sensitivity can be used to compute the sensitivity of a criterion related to the deformation shape, such as stress, or the relative displacement between two points on the structure.

The objective functions and constraint functions together with their sensitivities are supplied to an optimization algorithm. Use is made of a sequential quadratic programming (SQP) algorithm, following the ideas in [6]. We have chosen a fairly conservative optimization strategy, in order to reduce the likelihood of ending up in an infeasible and unstable design from which it is hard to recover.

OPTIMIZATION PROBLEM FORMULATION

This section discusses the performance objective function and the constraints.

Performance

The objective of the optimization is to improve closed-loop performance. While many metrics can be chosen to quantify closed-loop performance, we have chosen to look at the disturbance sensitivity transfer function $S(j\omega)$. This function is defined as [4]:

$$S(j\omega) = [I + L(j\omega)]^{-1},$$

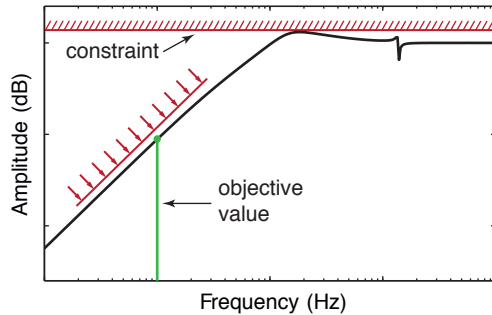


FIGURE 2. Usual shape of the disturbance sensitivity for one control loop. Attenuation is optimized at 10 Hz and amplification must remain below 6 dB everywhere.

in which $L(j\omega)$ is the loop transfer function. The objective is the magnitude of this function, computed at a frequency of 10 Hz, and averaged across the three control loops, i.e., the average of the diagonal elements of $|S(j\omega)|_{\omega=2\pi \cdot 10 \text{ Hz}}$ (cf. Fig. 2). Note that optional weightings could be assigned to the (in this case three) components. The advantage of the sensitivity functions is that these are dimensionless, giving a default normalization.

In addition to optimizing performance as just defined, a term is added to the objective which penalises the total mass of the structure. It may be that several structures exist that perform equally well, and in such cases a low-mass design is favoured.

Constraints

The following constraints are considered:

- The closed-loop is required to be stable;
- The diagonal elements of the disturbance sensitivity function $S(j\omega)$ are not allowed to exceed 6 dB in magnitude for robustness (cf. Fig. 2);
- During acceleration, the stage deforms due to the actuator and inertia forces. At maximum acceleration, the substrate layer may not deform by more than a certain amount to guarantee stability of the substrate. This requires the stage to have sufficient stiffness and is a mechanical constraint on the structure;
- In the diagram two sensors m_1 and m_2 are depicted. While these sensors are not part

of the control loops considered in this paper, they should be connected to the structure with a certain stiffness.

A final important consideration is that the dynamics seen by the control system, i.e., the transfer function $G(j\omega)$, depends on the sensor positions as the stage moves with respect to the metrology frame. To account for this fact, the design is performed simultaneously for three sensor positions (the center position and the extreme left/right positions of the stage). In each design iteration we optimize the worst-case performance out of three sensor locations, while ensuring that the constraints are satisfied for each of the three locations, effectively performing a robust design.

RESULTS

Initially, 2D problems have been considered in order to develop the optimization methodology and verify its feasibility. Several designs can be compared. The starting point for the optimization runs is a design in which the domain is filled with 75% density material (e.g., $E = 0.42E_0$ and $\rho = 0.95\rho_0$, following (1-2)). A result of the integrated approach in which controller and structure are optimized simultaneously is shown in Fig. 3d.

Next to the integrated optimization approach an eigenvalue optimization approach is performed. In this case the first natural frequency of the structure is maximized, subject to the constraint that the second natural frequency must be at least 10% higher than the first, to avoid that these frequencies cross each other. This result is shown in Fig. 3b.

For both designs, a black-and-white interpretation can be made by thresholding the grayscale designs (Figs. 3c and 3e). In this case a fairly arbitrary threshold value of 0.6 was chosen.

Each of the obtained designs, as well as their black-and-white interpretations can be compared to a completely solid design (Fig. 3a), in terms of performance and mass. The results are summarized in Figs. 4 to 6. Note that all data have been scaled for intellectual property reasons. The values obtained for the baseline design have been set to 100 and all other designs are related to those baseline values.

First, it appears from Fig. 4 that the integrated approach (d) yields the best average improvement

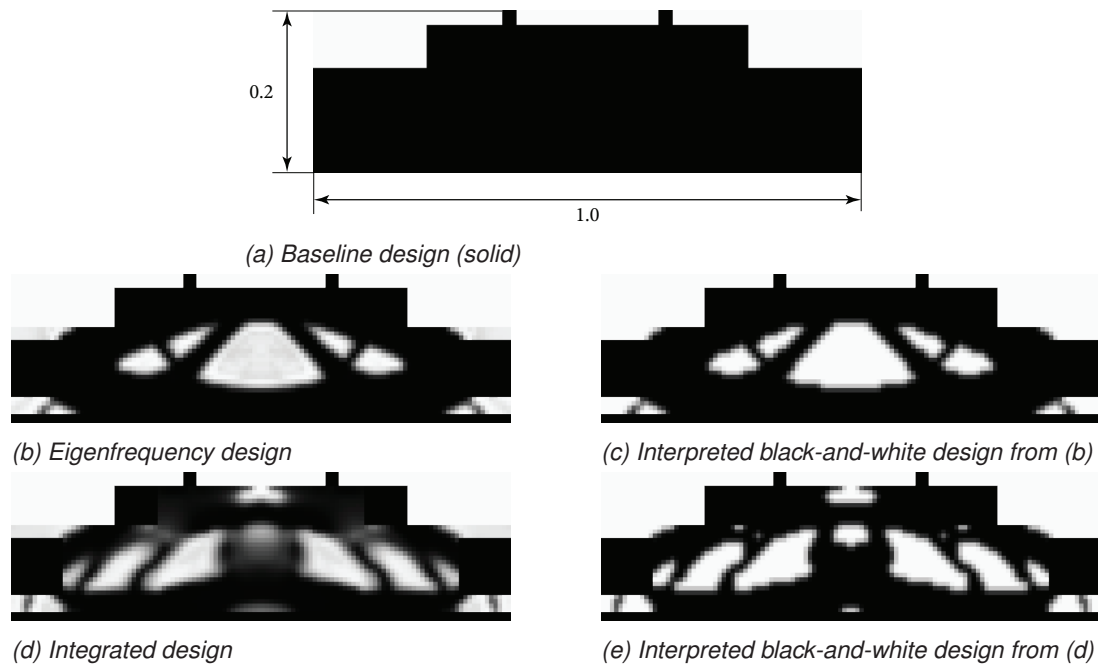


FIGURE 3. Example topologies comparing integrated and eigenfrequency designs, as well as their black-and-white interpretations. The performance figures are compared in Figures 4 to 6.

in disturbance attenuation. Second, from Fig. 6, it is clear that this performance is achieved while the first two natural frequencies are *reduced* by approximately 20%. So, in this case, it is not beneficial to raise the first natural frequency. This is because the first two modes are not bandwidth-limiting in the second and third control loops (respectively the y -position and θ -position) since they are collocated from the point-of-view of the respective control loops. Clearly, eigenfrequency design can only be performed with a good understanding of which subset of modes is relevant to overall performance. In fact, the eigenfrequency design (b) results in a lower average performance than the solid design while its lowest natural frequency is more than 10% higher than for the baseline design. Finally, the integrated approach also yields the lowest mass (Fig. 5), which is almost 30% below the solid stage mass.

The results obtained for this example demonstrate that non-trivial and fairly well-defined designs can be obtained, which could form a starting point for subsequent design. The integrated design (d) shown here has at least 20 dB better disturbance attenuation at 10 Hz in each of the motion DoFs while having smaller mass.

The algorithm relies on small changes in the plant dynamics, meaning that the nature of consecutive poles and zeros in the transfer functions does not change. It has difficulty when flexible modes start to cross each other. This may be less of an issue if the algorithm is applied to modify existing designs so that only small changes in the dynamics occur, but it needs attention to allow application to a wider set of problems.

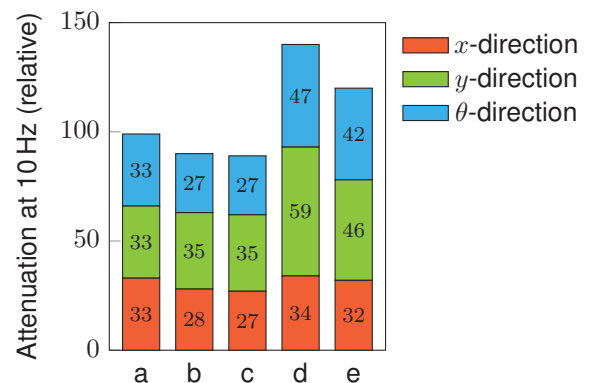


FIGURE 4. Improvement in the normalized disturbance attenuation figures compared to the baseline design (larger implies better attenuation). The letters a through e refer to the designs shown in Fig. 3.

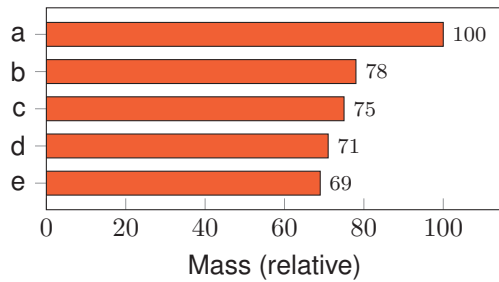


FIGURE 5. Mass reduction figures compared to the baseline design (smaller is lighter). The letters a through e refer to the designs shown in Fig. 3.

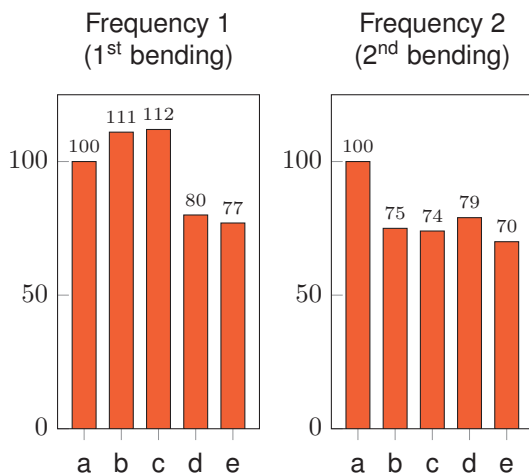


FIGURE 6. Natural frequencies (normalized) compared to the baseline design. The letters a through e refer to the designs shown in Fig. 3. Two modes are shown in Fig. 7.

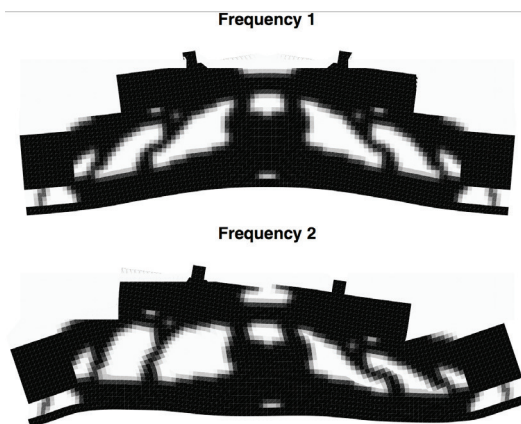


FIGURE 7. The first two mode shapes of design (e).

CONCLUDING REMARKS

In summary, it is shown that our methodology has the potential to become a useful tool for the design of motion stages in the precision industry and can deal with sophisticated objective functions. The main focus of subsequent research will be to incorporate more constraints originating from actual practice and to move towards 3D design. While our focus was on mechanical and control system specifications, ultimately, incorporating thermal specifications will be a necessity for precision applications.

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REFERENCES

- [1] Munnig-Schmidt R, Schitter G, Rankers A, van Eijk J. The Design of High Performance Mechatronics: High-tech Functionality by Multidisciplinary System Integration. 2nd ed. Delft University Press; 2014.
- [2] Bendsøe MP, Sigmund O. Topology Optimization: Theory, Methods and Applications. Engineering online library. Springer; 2003.
- [3] van der Veen G, Langelaar M, van Keulen F. Integrated topology and controller optimization of motion systems in the frequency domain. Structural and Multidisciplinary Optimization. 2015;51(3):673–685. Available from: <http://dx.doi.org/10.1007/s00158-014-1161-4>.
- [4] Skogestad S, Postlethwaite I. Multivariable feedback control, analysis and design. John Wiley and Sons; 1996.
- [5] Pedersen NL. Maximization of eigenvalues using topology optimization. Structural and Multidisciplinary Optimization. 2000;20(1):2–11.
- [6] Groenwold AA, Etman LFP, Wood DW. Approximated approximations for SAO. Structural and Multidisciplinary Optimization. 2010;41(1):39–56. Available from: <http://dx.doi.org/10.1007/s00158-009-0406-0>.

POLYMER DAMPER TECHNOLOGY FOR IMPROVED SYSTEM DYNAMICS

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INTRODUCTION

Requirements on the dynamic position stability of high-end mechatronic systems becomes increasingly tighter. This involves the moving systems, like stages, as well as the static systems, like force and metrology frames. A very effective and robust way to improve the system dynamics is by applying damping. This can either be done in an active or in a passive manner. In several applications, passive damping is more preferred than active. This mainly for reasons of robustness, applicability, performance and cost.

For our applications, several passive damping principles have been investigated (e.g. viscoelastic, viscous and eddy current damping). Viscoelastic damping is chosen for robustness, simplicity, amount of damping per material volume and design freedom of damper element.

Viscoelastic materials were applied in the past in various applications to achieve damping, however in many of these applications the precise performance was less critical and often by 'trial and error' the design had been realized. In our high-end applications, the complexity, requirements and conditions are such that extensive modelling is necessary and predictability of the final performance is crucial for a successful development of the entire system.

This paper describes the steps towards a predictable implementation of viscoelastic damping in systems to improve the system dynamics, starting with the theoretical background on viscoelasticity, followed by material characterization and the design process. Two examples of viscoelastic dampers are presented.

VISCOELASTIC BEHAVIOR

For viscoelastic materials, which exhibit damping, stress and strain are not in phase. The strain is lagging behind the stress by a certain phase angle δ . For dynamic modelling in the frequency domain, it is convenient to describe the linear viscoelastic behavior in complex

notation. Considering the complex time dependent strain as

$$\varepsilon^*(t) = \varepsilon_0 e^{j\omega t} \quad (1)$$

As a linear viscoelastic response we get for the corresponding stress

$$\sigma^*(t) = \sigma_0 e^{j(\omega t + \delta)} \quad (2)$$

It is common in the polymer technology domain, to describe the linear viscoelastic relation between stress and strain by means of the complex modulus E^* as

$$E^* = \frac{\sigma^*}{\varepsilon} \quad (3)$$

Substitution of (1) and (2) onto (3) and applying Euler's exponential relationship we get

$$E^* = \frac{\sigma_0}{\varepsilon_0} e^{j\delta} = \frac{\sigma_0}{\varepsilon_0} (\cos(\delta) + j \sin(\delta)) \quad (4)$$

We define two, in general frequency dependent material parameters, the storage-modulus E' and the loss-modulus E'' and rewrite (4) to

$$E^* = \frac{\sigma_0}{\varepsilon_0} (\cos(\delta) + j \sin(\delta)) = E' + jE'' \quad (5)$$

It is also common to describe the frequency dependent relation between stress and strain by means of 2 parameters, namely (1) the modulus of the complex modulus $|E^*|$ and by (2) the ratio between the loss-modulus and the storage-modulus called loss-factor η or $\tan(\delta)$.

$$|E^*| = \frac{\sigma_0}{\varepsilon_0} = \sqrt{E'^2 + E''^2} \quad (6)$$

$$\eta \equiv \frac{E''}{E'} = \tan(\delta) \quad (7)$$

And as such the complex modulus (5) is also given by $E^* = E' (1 + j\eta)$. In general the modulus and damping of polymeric materials (such as rubber) depends strongly on temperature and frequency [1], [2], [3]. A polymeric material

typically shows an enormous change in stiffness at the so-called glass transition temperature T_g (figure 1). With respect to damping, this glass transition appears to be the most important, due to the increase of the loss-modulus.

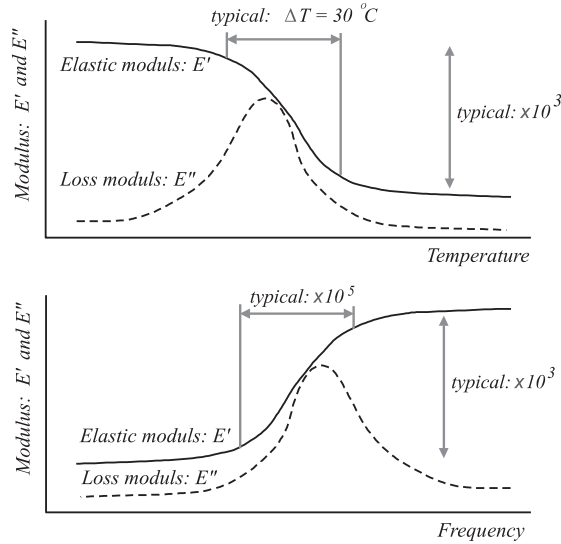


FIGURE 1. Effect of temperature and frequency on the different moduli (storage modulus E' , G' and loss modulus E'' , G'').

The change in stiffness is typically about 3 orders of magnitude over a temperature range of about 30°C. Typically, near the glass transition temperature, the ratio of loss modulus (E'' , G'') to storage modulus (E' , G') is 1:1 or even higher.

Next to temperature, the modulus of polymers also depends strongly on frequency. Qualitatively, the effect of frequency is the inverse of the effect of temperature (figure 1). This temperature to frequency relationship is described by the Williams-Landel-Ferry (WLF) equation [1].

In general, polymers show this transition temperature at substantial higher temperatures than room temperature (like hard plastics) or at substantial lower temperatures (like rubbers or elastomers). To get maximum damping at room temperature, proper blending of materials is necessary.

DYNAMIC VISCOELASTIC MODEL

A useful and common way to model the viscoelastic stiffness of polymers, required to model and predict the effect of damping on a system, is by means of combinations of elastic elements (ideal spring) and viscous elements (ideal

dampers). Figure 2 shows this multiple standard model, or Maxwell model.

$$k^* = k' + jk'' = k_0 + \sum_{i=1}^n \frac{j k_i d_i \omega}{j d_i \omega + k_i}$$

$$= \left(k_0 + \sum_{i=1}^n \frac{k_i d_i^2 \omega^2}{d_i^2 \omega^2 + k_i^2} \right) + j \left(\sum_{i=1}^n \frac{k_i^2 d_i \omega}{d_i^2 \omega^2 + k_i^2} \right)$$

FIGURE 2. Multiple standard model to describe the viscoelastic stiffness k^* .

This model can be used to describe the modulus in a general sense, i.e. Young's modulus E^* and shear modulus G^*

$$E^*(\omega) = E'(\omega) + jE''(\omega) \quad (9)$$

$$G^*(\omega) = G'(\omega) + jG''(\omega) \quad (10)$$

A fitting algorithm for the multiple standard model is applied to polymer EAR-C1105 [4], which is shown in figure 3.

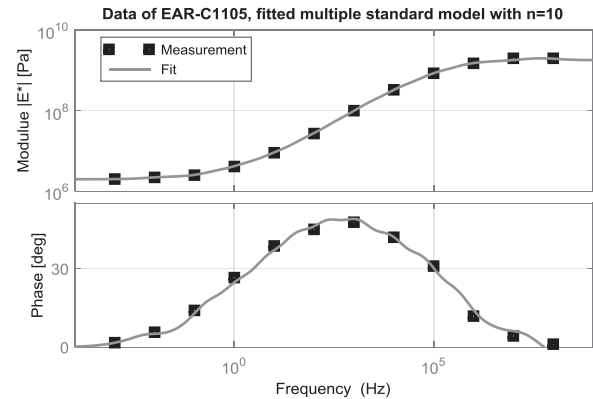


FIGURE 3. Modulus plot at 25°C of C1105 from [4] and multiple standard model fit with $n=10$.

MATERIAL CHARACTERIZATION

In general, for high-end mechatronic systems, an accurate material description over a large frequency range is necessary to predict the effect of damping properly. This requires an accurate measurement of the complex frequency dependent modulus (Young's and shear modulus) as function of frequency and temperature (e.g. $E^*(\omega, T)$).

To measure the dynamic modulus of viscoelastic materials, DMTA (Dynamic Mechanical Thermal Analysis) is commonly used [1]. Actually, the dynamic stiffness of a sample is measured over a limited frequency range at various temperatures $[T_1 \dots T_n]$. The WLF relation is used to interchange frequency and temperature to simulate the materials response over a large frequency range (typically 10 orders).

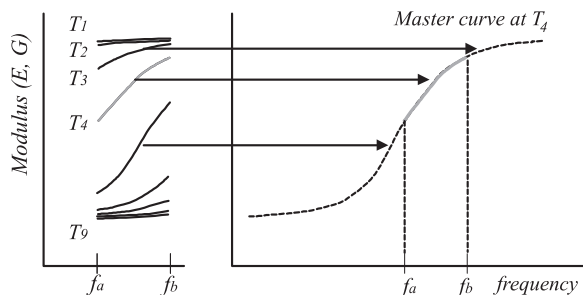


FIGURE 4. Master curve constructed from separate DMTA measurements at different temperatures. The master curve is obtained by shifting the individual measurement and is valid for a single temperature.

For our applications, the results of such a DMTA is found to be insufficiently accurate. Furthermore, such an apparatus is only feasible to handle predefined samples and is not able to characterize arbitrary damper designs. To overcome the limitations of the DMTA apparatus, two dedicated dynamic measurement setups have been built at MI-Partners. They are placed in a temperature controlled enclosure which can also be used to change the temperature during evaluation (typically 15...30°C).

The Damper Test Rig (DTR) has a generic interface which allows different samples up to 300x300x100 mm to be measured up to 400 Hz. The measurement principle is shown in figure 5.

The test apparatus generates a force by means of a direct drive motor. The generated force is measured by a load cell and the deformation of the sample is measured by 2 capacitive sensors (symmetrically along the force line). The dynamic Young's modulus can be determined by taking the sample geometry into account.

$$E^*(\omega, T) = k^*(\omega, T) / \psi \quad (11)$$

Where the constant ψ can be determined by a FEM analysis of the sample geometry.

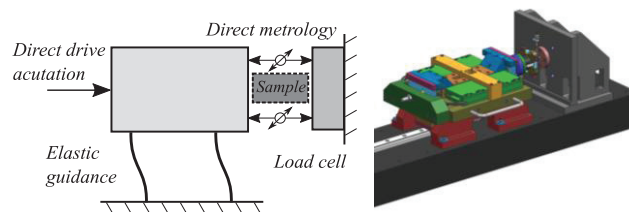


FIGURE 5. Principle (l) and overview (r) of the measurement Damper Test Rig (DTR), for material and damper characterization up to 400 Hz.

The High Frequency Damper Test Rig (HF-DTR) is equipped with accelerometers and a piezo load cell for their relatively flat amplitude and phase response at high frequencies. The force is generated by a modal exciter.

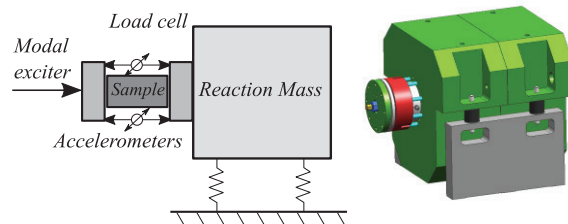


FIGURE 6. Principle (l) and overview (r) of the High Frequency Damper Test Rig (HF-DTR) for material characterization with dedicated sample geometry and damper characterization in the range of 50-5000 Hz.

The HF-DTR is limited in sample geometry and size, in order to measure from 50 Hz to 5 kHz with low uncertainties. To obtain a measurement uncertainty of the test within 5% over the full frequency range, the first internal dynamics in the measurement direction are designed above 9kHz. Furthermore corrections for the sensor dynamics (1) and the test setup dynamics (2) are required.

This includes (1) the correction for the (dynamic) accelerometer calibration data as provided by the sensor supplier; and (2) the correction for setup dynamics (inertia and compliance) which affect the measurement.

The test setup is verified by the measurement of an elastic reference sample, with constant magnitude and phase over frequency. This sample was calibrated in an external (static) test setup. Figure 7 shows the dynamic measurement of this elastic reference sample on the HF-DTR setup with and without and the effect of the sensor and test setup dynamics corrections.

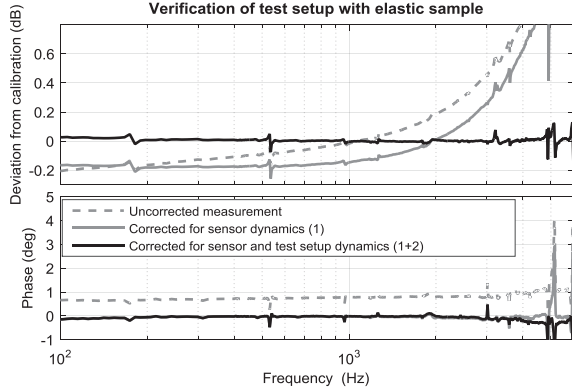


FIGURE 7. Verification of HF-DTR setup with an elastic reference sample and the effect of the sensor (1) and test setup dynamics corrections (2). The deviation from the calibration value is presented in dB.

MATERIAL DEVELOPMENT

To achieve the performance needed in our applications, dedicated viscoelastic materials have been developed in close cooperation with a supplier. These materials have been developed with specific properties, like high internal damping (>50%) at specific frequency ranges, suitable stiffness range, low outgassing characteristics for vacuum applications, high stability and repeatability of mechanical properties and good manufacturability.

DAMPER DESIGN

For various viscoelastic materials, the damping in shear and tension loading are the same (both load cases result in the same loss-factor), or

$$\eta(\omega) = \frac{E''(\omega)}{E'(\omega)} = \frac{G''(\omega)}{G'(\omega)} \quad (11)$$

Considering the very small strain levels, and the load case independent loss-factor, the material can be treated as homogenous isotropic linear viscoelastic, which can be characterized by e.g. the frequency dependent Young's modulus $E^*(\omega)$ and shear modulus $G^*(\omega)$. So the dynamic stiffness of an arbitrary specimen is given by

$$k^*(\omega, T) = \psi E^*(\omega, T) \quad (12)$$

The diagram in figure 8 shows the entire design cycle, including verification. The system dynamics, captured in a state space model, is imported in Matlab®. As such, complex systems can be evaluated containing all kind of dynamic characteristics, including feedback loops. Also the frequency dependent properties (stiffness and damping) of the viscoelastic material is

added in Matlab, by means of the above mentioned (dynamic) Maxwell models.

The damped dynamic system is evaluated and the properties (e.g. stiffness, position etc.) of the viscoelastic part can be optimized regarding the aimed system requirements. This process results in an optimal design target stiffness ($k_{optimal}^*$) for the geometrical design ($\psi_{optimal}$) of the damper module.

Accordingly the damper can be designed to the corresponding stiffness $k(\omega_0, T_0)$ at the (arbitrary) reference frequency and temperature, (e.g. $(\omega_0, T_0) = (25\text{Hz}, 22^\circ\text{C})$). The achieved design ψ_{actual} is analyzed by means of a homogenous isotropic linear elastic FE-model by applying $(|E^*(\omega_0, T_0)|, |G^*(\omega_0, T_0)|)$. Finally, the iteratively obtained actual geometry is evaluated in the dynamic system model (Matlab model).

Finally, the damper is produced in close cooperation with a supplier. The last step is to verify the damper properties, e.g. the geometry and dynamic stiffness, which can be measured in our test setups (DTR or HF-DTR)

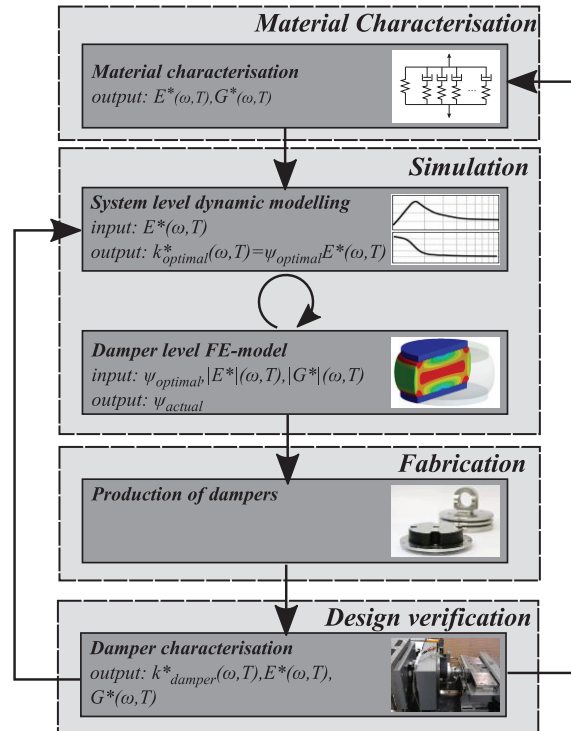


FIGURE 8. Damper design cycle including verification on damper level.

This complete design and realization cycle, from material characterization to design verification, can be realized within uncertainty of about 15% regarding the stiffness and damping of the damper at the target frequency. The variations include batch variations of the polymer, modeling and measurement inaccuracies. These small uncertainties allow for predictable and successful application of damping in high-end applications.

REALIZED DAMPER DESIGNS

Two examples of realized damper designs are presented below. The first example deals with a hexapod structure with suspension dynamics at lower frequencies, while the second one is used to damp structural dynamics above 1 kHz.

Optical module on hexapod structure

An optical module needs to be aligned relative to the optical system of a lithography tool. To perform the alignment adjustment over a relative large stroke, typical 50 mm, a hexapod configuration is used. By means of 6 spindles the optical module can be adjusted in all 6 degrees of freedom. The hexapod configuration is optimized such that the 6 suspension modes (i.e. first 6 eigenfrequencies of the system) are in a narrow band.

Beside the narrow band of suspension frequencies, to avoid interference with other heavy modules and to minimize excitation by main disturbance sources, also damping of the suspension modes is necessary. The required amount of modal damping should exceed a ratio of 10%. To apply damping to the suspension modes, a dedicated polymer damper has been developed.

Each spindle is provided with a damper unit attached to the bottom of the spindle (figure 9). An elastic spring parallel to the damper module supports the static load while preventing creep, to guarantee the positioning stability of the optical module. In this configuration the damper is sealed by the metal housing of the support spring, this to comply with the tight outgassing requirements as dictated by the clean room specifications.

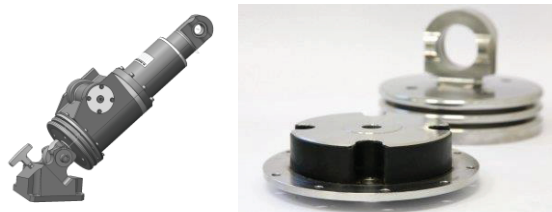


FIGURE 9. Damper module attached to the bottom of the spindle with the polymer damper element inside.

The dampers have been realized and the damping on system level has been verified by modal measurement on the entire structure. Based on the dynamic model in Matlab-Simulink, proper actuation and measurement point have been selected.

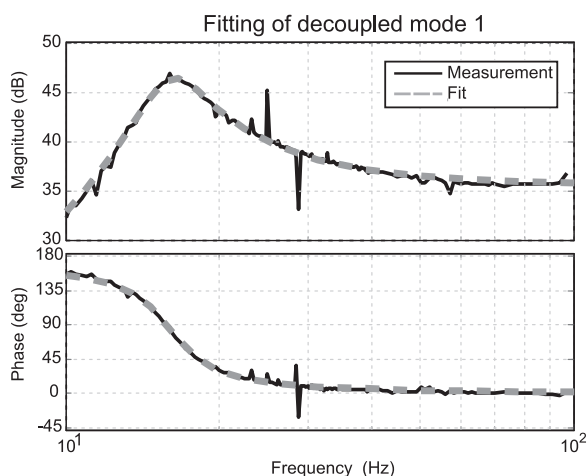


FIGURE 10. Measured and fitted transfer functions of the first (decoupled) suspension mode (force to acceleration).

To increase the quality of the measurements, a modal exciter was used instead of hammer excitation. To evaluate the modal damping values, the 6 suspension modes need to be separated (decoupled). Due to the high damping, separation could not be based properly on frequency only. The Matlab-Simulink model was used to determine the modal decoupling matrices. Applying this decoupling on the experimental results yields a proper modal decoupling with dominant diagonal terms and minor off-diagonal terms. Onto the corresponding decoupled frequency response data (e.g. figure 10), second-order models are fitted to determine damping. The frequency response function of the first suspension mode

is depicted in figure 10, showing a modal eigenfrequency of about 15Hz with 15% relative modal damping. The difference between the modelled damping and the actual damping on system level was smaller than 18% for all 6 suspension modes.

Damping of positioning module

For highly dynamic stages (e.g. fast tool servo, positioning stage) the servo bandwidth is often limited by internal mechanical resonance frequencies. In our case, in order to increase the controller bandwidth, a dedicated damper is designed to damp these internal resonances.

Results of measurements and simulations of an undamped highly dynamic positioning system are shown in figure 11. The first internal modes are near 2 kHz and appear as highly undamped resonance peaks in the frequency response function, thereby limiting the controller bandwidth.

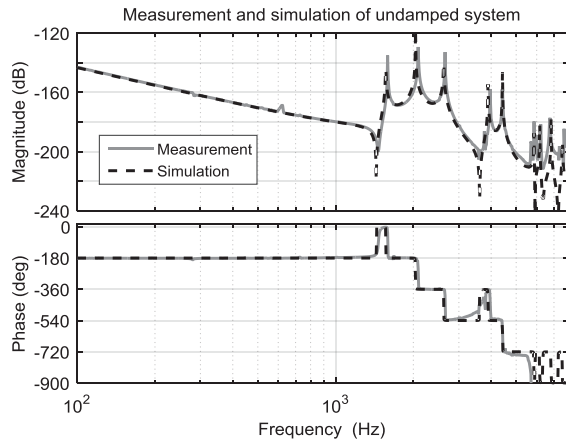


FIGURE 11. Measurement and simulation of a highly dynamic positioning stage without damping.

A damper has been optimized to damp the system internal modes. By applying the viscoelastic based damping (figure 12), the dominant 3 resonances are suppressed by about 30 dB or more. This allows to increase the controller bandwidth by roughly 20-30%, next to an enormous increase in system robustness. This application shows the high potential of the application of viscoelastic materials in high-end mechatronic system.

The bode plot shows the measured and the simulated frequency response function, which are in good correspondence. The deviation between the simulated and measured frequency

response at the first 3 damped resonances is within 2 dB, which shows the high predictability of this technology.

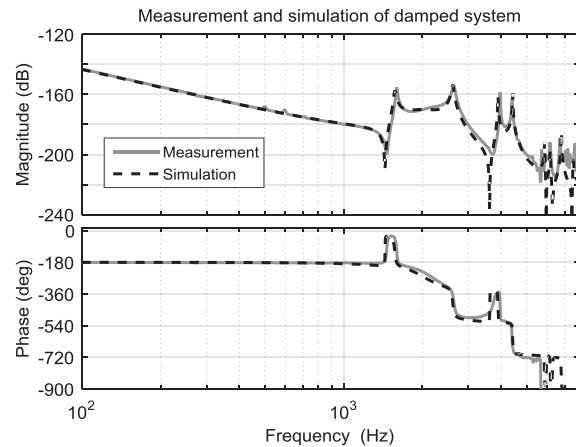


FIGURE 12. Measurement and simulation of a highly dynamic positioning stage with viscoelastic damping. The main resonances are reduced by 30 dB or more.

REFERENCES

- [1] Corsaro, R.D.; Sperling, L.H.: *Sound and Vibration Damping with Polymers*. In: Proceedings 197th National Meeting of the ACS. April 1989 Dallas. ACS Symposium Series. ISBN 0-8412-1778-5. S. 5-22.
- [2] Jones, D.L.G.: *Handbook of Viscoelastic Vibration Damping*. Chichester: John Wiley & Sons Ltd, 2001.
- [3] Rivin, E.: *Passive Vibration Isolation*. New York: ASME Press, 2003.
- [4] EAR Special Composites (www.earsc.com).

A NOVEL APPROACH FOR REDUCING THE SETTLING TIME OF ROLLER BEARING NANOPositionING STAGES USING HIGH FREQUENCY VIBRATION

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OVERVIEW

This paper presents vibration assisted nanopositioning (VAN) – a novel approach for mitigating the adverse effects of pre-rolling friction on the settling time of roller bearing nanopositioning stages using high frequency vibration. The novelty of VAN is in the way it synergistically combines the mechanical design and control of the stage to mitigate pre-rolling friction without jeopardizing positioning precision. After a brief introduction, the concept of VAN is described and its superior performance and robustness relative to a recently-proposed friction compensation method is demonstrated in simulations. The design of a prototype VAN stage is then outlined, and the prototype stage is used in preliminary experiments to demonstrate up to 61.7% reduction of settling time compared to a stage without VAN.

BACKGROUND AND MOTIVATION

Nanopositioning (NP) stages are used for precise positioning in a wide range of ultra-precision processes, ranging from MEMS inspection to micro additive manufacturing. They can be constructed using flexure, fluidic, magnetic or roller bearings. Flexure-based stages are compact, low-cost and friction free. However, their limited stroke and load capacity make them unsuitable for large-range, medium-to-high-payload applications e.g., nanoimprint lithography [1]. Magnetic bearings are capable of large range nanopositioning but their commercial use is currently relegated to the highest end stages (such as those used in semiconductor manufacturing [2]) because of their high costs and complexities. Fluidic (i.e., hydrostatic or aerostatic) bearings are also capable of large-range nanopositioning but they are not suitable for clean room and/or high vacuum environments.

Roller bearing NP (RB-NP) (i.e., crossed roller and linear ball bearing) stages are the most cost effective type of NP stages. Moreover, they are currently the only commercially viable option for a growing number of large-range NP applications that must be performed in high or ultra-high vacuum environments, e.g., scanning electron microscopy [3] and focused ion beam [4]. However, the presence of friction adversely affects their performance. A typical RB-NP stage takes a very long time to reach its target position in point-to-point positioning due to the so-called “pre-rolling friction,” which dominates as the stage gets within microns of its target position [5]. Such long settling times severely hamper the throughput of the processes for which RB-NP stages are used.

The state of art in addressing friction in RB-NP stages is to perform model-based friction compensation through feed forward or feedback controllers [6]. Friction compensation works well if the friction model is sufficiently accurate [6]. However, due to the extreme variability of friction dynamics, particularly in the pre-rolling regime, such methods typically suffer from poor robustness. Several methods have been proposed in the literature for improving the robustness of friction compensation, mainly through model parameter adaptation [6, 7]. However, the convergence of adaptation schemes is often difficult and slow because the identification signals are not rich (or persistent) enough [6]. Another approach is to use a feedback controller with very good rejection of un-modeled or poorly modeled disturbances [6]. The result is often a high gain or nonlinear controller which could lead to instability or undesirable limit cycles.

Dither is well-known to be an effective and robust way of mitigating nonlinear phenomena like friction, backlash and hysteresis [8]. For dither to work effectively, the frequency and

amplitude of oscillations must be high [8]. Therefore, when considering dither for friction mitigation in RB-NP stages, the following problems emerge:

- Traditionally, the dithering force (F_d) is applied indirectly to the location of friction (F_f) by adding it to the servo actuation force (F_a) of the stage (see Figure 1(a)). However, the effectiveness of dither is greatly attenuated by the low-pass filtering effect resulting from the stage dynamics.
- Applying high amplitude dither to a precise positioning stage through its servo actuator causes excessive vibration of the stage thus jeopardizing its positioning performance; there is no guarantee that the stage will stop at the desired position after dither is turned off.
- When high amplitude dither is maintained for prolonged periods of time, it causes accelerated wear of mechanical components and excessive heat generation.

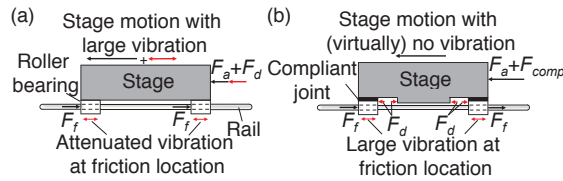


FIGURE 1 Schematic representation of (a) traditional approach; and (b) proposed VAN approach of applying dither to a positioning stage.

VIBRATION ASSISTED NANOPositionING

Concept

To address the limitations of traditional dithering, we propose a novel approach of applying dither to RB-NP stages which we call vibration assisted nanopositioning (VAN). As shown in Figure 1(b), it comprises the following features:

- Each roller bearing is not rigidly attached to the stage; rather it is attached using a joint that is compliant in the stage's direction of motion but stiff in other directions.
- F_d is applied directly to each roller bearing using a small actuator (e.g., voice coil or piezo stack) to create large enough vibration amplitudes at the location of friction.
- F_d is applied with a phase difference of 180° to the bearings on opposite ends of the stage

such that the reaction forces transmitted to the stage (mostly) cancel out.

- The adverse effects of un-cancelled vibration, heat and wear are minimized by regulating the applied dither force and/or by applying a compensating force (F_{comp}) through the actuator of the stage.

Simple Model of VAN Stage

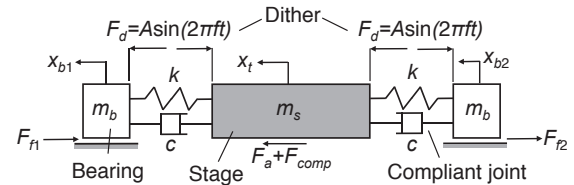


FIGURE 2 Simple model of VAN stage.

Figure 2 shows a simple three-mass model of a closed-loop controlled VAN stage. The compliant joint connecting the stage of mass m_s to each bearing of mass m_b is modeled by spring stiffness k and damping coefficient c . Dither is applied in the form of a harmonic excitation force F_d , with amplitude A and frequency f . The system dynamics are described by

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{C}\dot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{F}, \quad (1)$$

where $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ are respectively the mass, damping and stiffness matrices, while $\mathbf{u}$ and $\mathbf{F}$ are respectively the displacement and force vectors of the system. They are given by

$$\mathbf{M} = \begin{bmatrix} m_b & 0 & 0 \\ 0 & m_b & 0 \\ 0 & 0 & m_s \end{bmatrix}, \quad \mathbf{C} = \begin{bmatrix} c & 0 & -c \\ 0 & c & -c \\ -c & -c & 2c \end{bmatrix},$$

$$\mathbf{K} = \begin{bmatrix} k & 0 & -k \\ 0 & k & -k \\ -k & -k & 2k \end{bmatrix}, \quad \mathbf{F} = \begin{bmatrix} F_d - F_{f1} \\ -F_d - F_{f2} \\ F_a \end{bmatrix}, \quad \mathbf{u} = \begin{bmatrix} x_{b1} \\ x_{b2} \\ x_s \end{bmatrix}, \quad (2)$$

where x_b and x_s are respectively the displacements of stage and bearings. The subscripts 1 and 2 attached to F_f and x_b are used to distinguish between the displacements and friction forces of each bearing, which may be different.

SIMULATION BASED ANALYSIS OF VAN

VAN is evaluated in simulations by comparing its settling performance with that of the Nonlinear Integral Action Settling Algorithm (or NIASA for short) proposed by Bucci et al. [9]. In [9], a PID

controlled Aerotech ALS-130H RB-NP stage with total moving mass $m = 1.5$ kg is used to evaluate NIASA in simulations. Its P and D gains are respectively 0.8 N/ μm and 8.5×10^{-4} N·s/ μm , but its integral gain is a nonlinear function of the friction force F_f , assumed to be described by the Dahl model [10], given by

$$\frac{dF_f}{dx_r} = \sigma \left(1 - \frac{F_f}{F_c} \text{sgn} \dot{x}_r \right)^i, \quad (3)$$

where x_r is the relative displacement between the sliding surfaces, $\sigma = 8$ N/ μm is the initial stiffness, $F_c = 1$ N is a measure of the Coulomb friction force and $i = 1$ is a shape factor.

To compare with NIASA, a PID controlled VAN stage is modeled as shown in Figure 2, with $m_s = 1$ kg, $m_b = 0.25$ kg, $k = 5$ N/ μm , $c = 2 \times 10^{-5}$ N·s/ μm (corresponding to 1% damping ratio) and $F_{f1} = F_{f2} = F_f$. Note that the total mass of the VAN stage (i.e., $m = m_s + 2m_b$), its P and D controller gains, and its frictional force parameters are exactly the same as used for NIASA in [9], as described in the preceding paragraph. A regular integral gain of 1 N/(s· μm) is used for VAN (as opposed to the nonlinear integral action used in NIASA).

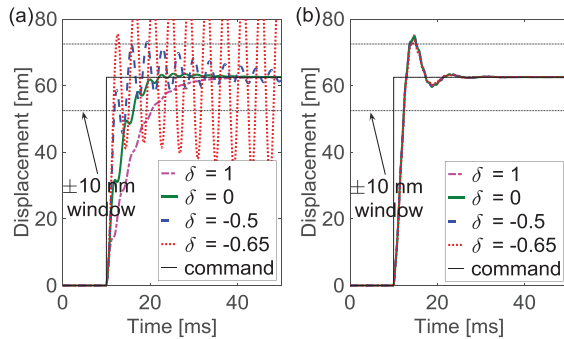


FIGURE 3. Settling results of (a) NIASA and (b) VAN in response to a 62.5 nm step command using different values of initial contact stiffness in the Dahl model. Note that $A = 3$ N and $f = 500$ Hz for VAN.

Table 1. Comparison of settling times of NIASA and VAN into the ± 10 nm window shown in Figure 3 for various values of δ .

Method	Settling Time [ms]			
	$\delta = 1$	$\delta = 0$	$\delta = -0.5$	$\delta = -6.5$
NIASA	10.3	6.2	9.5	N/A
VAN	5.2	5.2	5.1	5.1

Figure 3 compares the settling responses of NIASA and VAN to a 62.5 nm step command used by Bucci et al. [9]; a ± 10 nm window is used as the settling window. A multiplicative uncertainty parameter $\delta \in [-1, \infty)$ is introduced by Bucci et al. [9] into the contact stiffness σ of the Dahl model described by Eq. (3) to help demonstrate the robustness of NIASA to changing friction. It is observed that NIASA is indeed stable within the parameter band $\delta \in [-0.5, 1]$ that is presented in [9]. However, its settling performance varies significantly as δ deviates from its nominal value of zero as shown in Figure 3(a). Notice that the system with NIASA experiences severe oscillations for $\delta = -0.5$, and suffers from a limit cycle with large amplitude for $\delta = -0.65$, causing it to not settle. On the other hand, as shown in Figure 3(b), VAN (with $A = 3$ N and $f = 500$ Hz) demonstrates remarkable robustness and settling performance in the presence of varying friction.

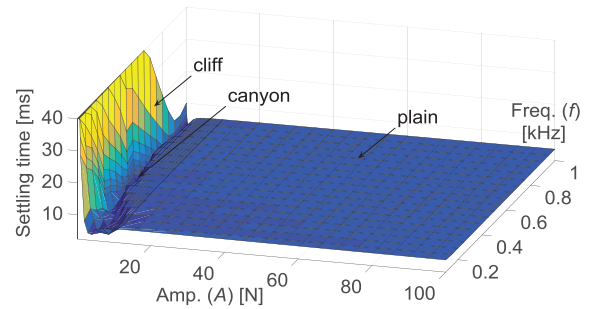


FIGURE 4. Effect of dither amplitude (A) and frequency (f) on settling time for VAN stage

It is of interest to study the effect of A and f on the settling performance of VAN. Figure 4 shows the settling times of VAN as functions of A and f for the same test case as in Figure 3 (with $\delta = 0$). The settling time characteristics can be said to have a cliff, a canyon and a plain. At very low amplitudes and frequencies, the addition of dither produces little or no effect in reducing settling time. But beyond a certain critical frequency, there is a sudden drop in settling time over the cliff to its lowest levels (i.e., the canyon). It is observed that this critical frequency mainly depends on A . As A (and to some extent f) is increased further, the settling time increases slightly after which it remains largely unchanged (i.e., the plain). In terms of robustness, the plain is excellent because it provides a wide range of A and f values to choose from without losing performance. The problem is that the plain occurs at relatively higher A or f values, which could pose

challenges with regard to heat generation and bearing wear. The canyon provides optimal settling times at relatively lower A and f values than the plain. Therefore, operating within the canyon region could provide huge benefits with regard to the reduction of heat and wear. However, the challenge is that the canyon is very narrow and close to the cliff; settling times could rise sharply (up to 5 times in Figure 4) if the critical amplitude or frequency is crossed.

In the results presented in Figures 3(b) and 4, because $F_{r1} = F_{r2} = F_f$ is assumed, the reaction forces due to dithering perfectly cancel out, leading to zero vibration at the stage. If $F_{r1} \neq F_{r2}$, unequal reaction forces are generated and the stage experiences continuous vibration which may cause it to not settle. As discussed in the previous section, such unwanted vibration can be mitigated by regulating the applied dither and/or by injecting a compensating force F_{comp} via the stage's servo controller. In this preliminary work, we evaluate a simple regulation technique where dither is applied and then suddenly turned off after duration T , starting when the reference command reaches the target position. Such an approach is suitable for point-to-point positioning tasks where high precision is only required after the target position is reached.

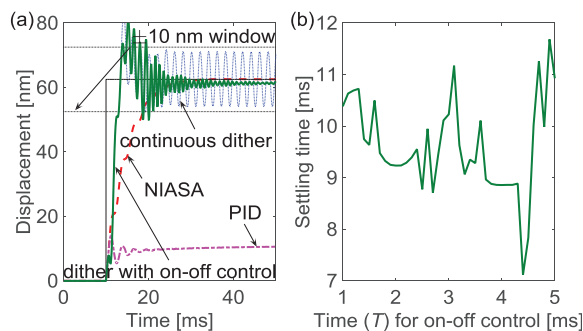


FIGURE 5. (a) Settling results of VAN in response to a 62.5 nm step command with imperfect cancellation of dithering reaction forces (compare with Figure 3(b)); (b) effect of time (T) on settling time for VAN

Let us re-consider the scenario shown in Figure 3(b) for VAN, but with $\delta = 1$ for F_{r1} and $\delta = 0$ for F_{r2} . Figure 5(a) compares the settling behavior of NIASA and VAN with and without on-off dither control to that of a regular PID controlled stage. Notice that without on-off dither control, VAN stage settles into the ± 10 nm window in 9.3 ms while NIASA settles in 8.7 ms, which are both

huge improvements compared to the regular PID which does not settle within the simulation time of 50 ms. However, because of the imperfect cancellation of reaction forces, VAN continues to vibrate as long as dither is maintained. The lingering vibration can be eliminated by shutting off dither as shown in the figure for $T = 3$ ms. Notice, however, that on-off control introduces some transients before the vibration dies down completely. The transient vibration could be reduced by making the on-off control less abrupt or if the stage has extra damping (as will be seen later in the experiments).

Figure 5(b) shows the settling times of VAN with on-off control for various values of T ranging from 1 to 5 ms. The observed fluctuation in settling time as T changes is partly due to the aforementioned transient vibration as well as the fact that the stage is at different positions (relative to the target position) when dither is turned off. Nonetheless, the deformed springs provide extra forces to overcome pre-rolling friction after dither is turned off, leading to much faster settling of the stage to the target position compared to the regular PID.

PROTOTYPE DESIGN AND EXPERIMENTS

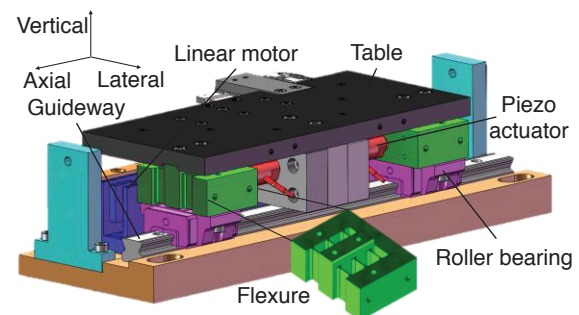


FIGURE 6. CAD model of a prototype VAN stage.

A prototype VAN stage is designed according to the general concept described in the previous section. Figure 6 shows the CAD drawing of the prototype, which is designed to have a moving mass of about 2 kg and a travel range of 50 mm. The stage is guided by a set of high-rigidity radial type linear ball bearings, riding on a super-precision grade rail. An ironless linear motor with 23 N continuous force is selected to drive the stage. The table position is measured using a 4.88 nm-resolution linear encoder. A pair of preloaded piezo actuators is selected to provide the dither force; each actuator has a

travel range and blocking force of up to 15 μm and 300 N, respectively, with a maximum operating frequency of 6000 Hz.

A simple flexure hinge, made of AISI 304 stainless steel, is used to connect the table to each of the roller bearing. Its role is to provide sufficient compliance in the direction of dithering while remaining rigid in other directions. Table 2 compares its stiffness values in different directions (obtained using SolidWorks® finite element analysis) with those of the linear bearing. With an axial rigidity of 6.9 N/ μm , the flexure can provide around 11 μm of displacement (with 77 N force) at the maximum operating voltage of the piezo actuators (i.e., 100 V). Notice from Table 2 that the vertical and lateral stiffness of the flexure are much higher than its axial stiffness. As a result, the combined stiffness of the flexure and linear bearing in the vertical and lateral directions have same order of magnitude as those of the linear bearing alone. It is important to note that at an applied load of 77 N, the highest maximum von Mises stress of the flexure is 43 MPa, which is much lower than the endurance limit (203 MPa) of 304 stainless steel. So, even at its worst-case loading, infinite fatigue life is guaranteed for the flexure.

Table 2. Stiffness of linear ball bearing and flexure (N/ μm).

	Axial	Vertical	Lateral
Bearing	N/A	175	58.3
Flexure	6.9	310	331
Combined	6.9	111.9	49.6

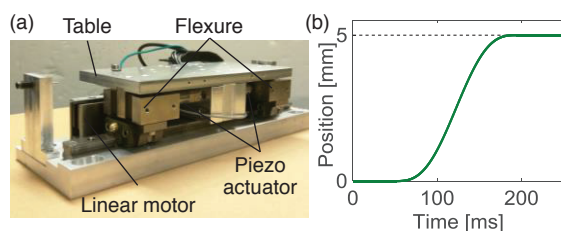


FIGURE 7. (a) Assembled VAN stage; (b) commanded position of a typical 5 mm move-and-settle operation.

Figure 7(a) shows an in-house-built prototype of the VAN stage. A typical 5 mm “move-and-settle” profile is used as the reference trajectory as shown in Figure 7(b). It has maximum acceleration/deceleration of 1500 mm/s² with a maximum velocity of approximately 70 mm/s. Only the settling portion, after the target position is reached, is considered in this paper. A PID

controller with velocity and acceleration feed forward is used to control the stage. The PID controller is tuned to achieve a closed loop bandwidth of about 195 Hz. It is implemented using a real time controller (dSPACE, DS1007) with current commands sent through a PWM amplifier. The two piezo actuators are controlled in open loop by sending sinusoidal commands through their voltage amplifiers.

The friction behavior of the prototype stage has been found to not conform to the Dahl model, so NIASA is not a suitable benchmark for experiments. Therefore, the performance of VAN is evaluated in experiments by studying the settling behavior of the prototype stage with and without dither. Figure 8 compares the settling performance of the stage into a 100 nm window with and without dither. Without dither, the stage takes 55.1 ms to settle. Dither with peak-to-peak amplitude of 20 V (corresponding to 15 N) at 500 Hz is added and, as expected, the stage experiences steady state vibrations due to imperfect cancellation of the reaction forces. The imperfect cancellation of reaction forces is caused by non-idealities in the stage, like differing friction forces on each bearing and slight differences in the performance of each piezo actuator and amplifier. The on-off control used in the simulations is applied (with $T = 7.5$ ms), causing the stage with dither to settle within 19.4 ms, which is 64.8% shorter than the case without dither. Notice that no significant transients are generated by the abrupt switching; this is in part due to the higher damping provided by the roller bearings, compared to the model used in simulations.

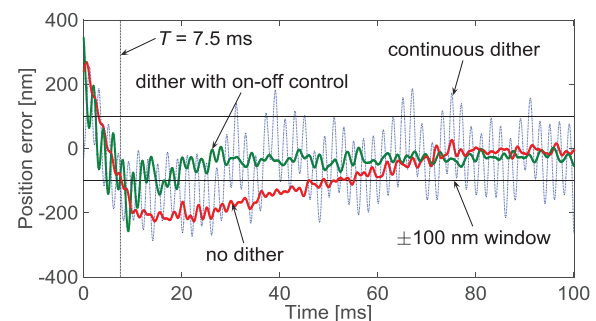


FIGURE 8. Typical settling response of no dither, continuous dither and dither with on-off control ($T = 7.5$ ms).

Figure 9 plots the settling performance of the prototype stage with on-off dither control for various values of T , in comparison with the case without dither. It is observed that with and

without dither, the settling performance of the stage varies a bit, due to differences in friction from position to position along the travel of the stage. The plots in Figure 9 show the mean settling times based on 20 trials at random positions of the table within its travel range; the error bars indicate one standard deviation. Without dithering, the mean settling time is 53.3 ms with a standard deviation of 4.8 ms, which shows the slow settling performance in comparison with the length of the reference trajectory (i.e., 100 ms). With on-off control, the stage with dither is able to achieve a mean settling time of 20.4 ms at $T = 7.5$ ms (with a standard deviation of 4.1 ms), which is 61.7% less than the case without dither. Apart from this optimal value, notice that with dither the system performs consistently better than the no-dither case for all values of T . Moreover, the proposed method is robust, considering that the stage is at random positions when dither is turned off for each trial.

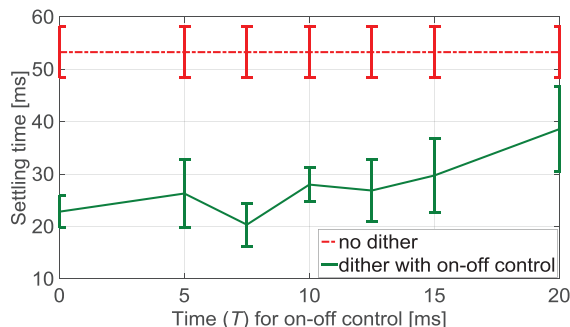


FIGURE 9. Comparison of mean settling times into the ± 100 nm window for no dither and on-off controlled dither with different T

CONCLUSIONS AND FUTURE WORK

In this paper, a novel approach for improving settling performance of roller bearing nanopositioning stages using high-frequency vibration (or dither) has been presented; it is called vibration assisted nanopositioning (VAN). It has been shown using simulations that VAN achieves superior performance and robustness compared to a model based friction compensation. A prototype VAN stage has been designed and built. Preliminary experiments conducted using the designed prototype (with a rudimentary on-off dither control technique) have demonstrated up to 61.7% reduction in mean settling time in point-to-point positioning. Future work will investigate advanced dither and stage control techniques, based on rigorous theoretical

analyses, to help optimize the performance and robustness of VAN.

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REFERENCES

- [1] Lan H, Ding Y, Liu H, Lu B. Review of the Wafer Stage for Nanoimprint Lithography. *Microelectron. Eng.* 2007; 84: 684–688.
- [2] Kim W, Trumper DL. High-precision Magnetic Levitation Stage for Photolithography. *Precis. Eng.* 1998; 22: 66–77.
- [3] Sitti M. Survey of Nanomanipulation Systems. *Proc. 2001 1st IEEE Conf. Nanotechnology.* 2001; 75–80.
- [4] Melngailis J. Focused Ion Beam Technology and Applications. *J. Vac. Sci. Technol. B: Microelectron. Nanom. Struct.* 1987; 5: 469.
- [5] Maeda Y, Iwasaki M. Rolling Friction Model-based Analyses and Compensation for Slow Settling Response in Precise Positioning. *IEEE Trans. Ind. Electron.* 2013; 60: 5841–5853.
- [6] Armstrong-Hélouvry B, Dupont P, De Wit CC. A Survey of Models, Analysis Tools and Compensation Methods for the Control of Machines with Friction. *Automatica.* 1994; 30: 1083–1138.
- [7] De Wit CC, Lischinsky P. Adaptive Friction Compensation with Partially Known Dynamic Friction Model. *Int. J. Adapt. Control Signal Process.* 1997; 11: 65–80.
- [8] Thomsen JJ. Some General Effects of Strong High-frequency Excitation: Stiffening, Biasing and Smoothing. *J. Sound. Vib.* 2002; 253: 807–831.
- [9] Bucci BA, Cole DG, Ludwick SJ, Viperman JS. Nonlinear Control Algorithm for Improving Settling Time in Systems with Friction. *IEEE Trans. Control Syst. Technol.* 2013; 21: 1365–1373.
- [10] Dahl P. A Solid Friction Model. *Aerosp. Corp. El. Segundo, CA.* 1968; 158.

ACTIVE ASSIST DEVICE FOR HEAT AND VIBRATION REDUCTION IN PRECISION SCANNING STAGES

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OVERVIEW

In prior work [1], the authors have presented a novel magnet assisted stage concept that simultaneously reduces the heat and vibration of precision scanning stages, due to the high accelerations and decelerations (acc/dec) of moving parts. However, the stage in [1] was configured with passive assist devices (PADs) [2], which limited its ability to provide assist for scan trajectories involving variable scan strokes/positions. To address this limitation, this paper presents an approach for configuring the stage with active assist devices (AADs). The superior performance of the AADs in reducing heat and vibration for scan trajectories with variable scan strokes/positions is experimentally demonstrated using two practical case studies related to silicon wafer scanning. Compared to the PAD configuration, over 77% and 52% reductions in heat and vibration, respectively, are achieved.

INTRODUCTION

Precision scanning stages provide repetitive motions required in a variety of advanced manufacturing processes like laser patterning, 3D printing, hard drive manufacturing, and silicon wafer lithography and inspection. Each scanning stroke consists of one constant velocity (CV) and two motion reversal (MR) regions, one at each end of the CV region. The CV region is where the actual manufacturing process (e.g., lithography or inspection) takes place, so positioning must be extremely precise. The MR regions are not useful to the manufacturing process; therefore, they must be executed with high acc/dec to boost throughput while ensuring that the precision of the CV regions is not compromised. The resulting high inertial forces that are borne by the linear motor actuators cause Joule heating, proportional to the square of the motor current, leading to increased thermal errors. They also cause residual vibration of the isolated machine base, which adversely affects positioning speed and precision.

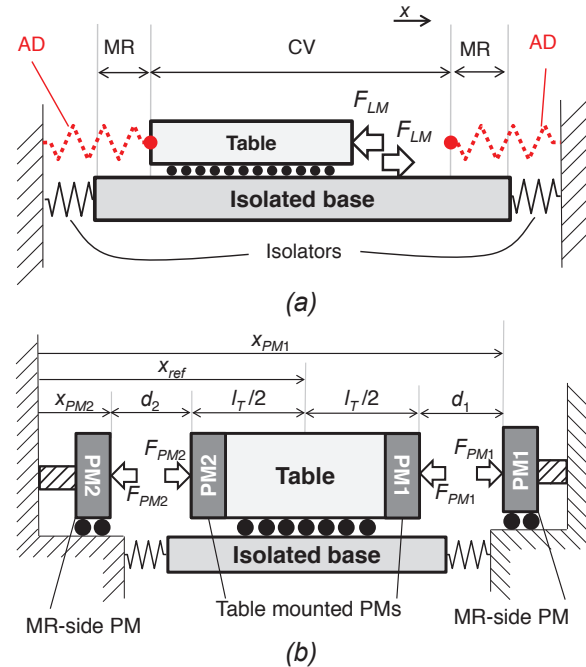


FIGURE 1. (a) Schematic of scan axis of precision scanning stage including assist devices (ADs). (b) Realization of ADs using magnetic repulsion. Movable PMs channel reaction forces to ground.

The authors have recently presented a magnet assisted stage concept, shown in Figure 1, for simultaneously reducing heat and vibration during scanning. It consists of two assist devices (ADs) in the form of springs at either side of the scanning table, designed using a pair of repelling permanent magnets (PMs). The ADs provide regenerative braking of the table when they are inside the MR regions, thus reducing the linear motor force, F_{LM} , needed for acc/dec. Vibration is reduced by channeling the PM assist reaction forces, F_{PMi} , from the ADs to the rigid ground, where $i = \{1, 2\}$. The nonlinear force-distance relationship of repelling PMs allows the ADs to be effectively detached from the table once the table leaves the MR regions. This way, the ADs do not transmit ground vibration into the CV region during precision scanning. In [1], the

positions x_{PMi} of the MR-side PMs were configured to be fixed, even if optimized, for a given scan trajectory, x_{ref} , of the table; in other words, the PM assist devices were configured as PADs. However, this approach limits versatility and performance of the PM assist devices in reducing heat and vibration for scan trajectories involving variable scanning strokes /positions. In this paper, x_{PMi} are not fixed, but allowed to vary freely when executing a scan trajectory, thus converting the PADs in [1] to AADs. A method for generating x_{PMi} to maximize assist (i.e., minimize F_{LM}), while keeping to the kinematic limits imposed on x_{PMi} due to actuator limitations, is presented and validated.

MAGNET ASSISTED STAGE

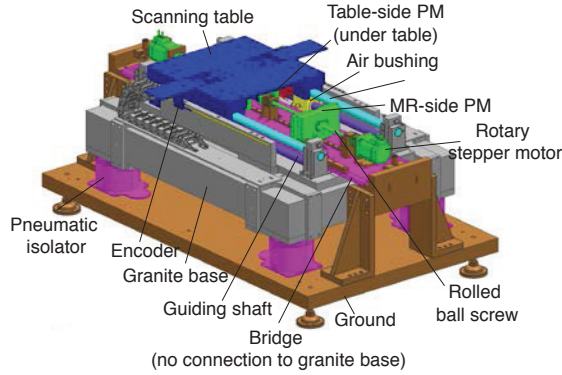


FIGURE 2. CAD model of magnet assisted stage reported in [1].

The CAD model of prototype magnet assisted stage presented in [1] is shown in Figure 2, to provide some context for this paper; its performance specifications are summarized in Table 1. Two pairs of PMs (i.e., PM1 and PM2) are installed on either end of the stage. An inexpensive servo system, driven by a stepper motor, is employed to automatically adjust x_{PMi} as shown in Figure 1 (b). Notice from Figure 2 that the bridge, upon which the servo systems for the MR-side PMs are mounted, does not contact the granite base, thus allowing the reaction of F_{PMi} to be conducted to the ground without disturbing the base. The speed and acceleration limits of each MR-side PM are $v_{PM,max} = 0.15$ m/s and $a_{PM,max} = 10$ m/s², respectively. The distance d_i between each PMi pair are given by

$$d_1(t) = x_{PM1}(t) - x_{ref}(t) - l_T / 2 \quad (1)$$

$$d_2(t) = x_{ref}(t) - x_{PM2}(t) - l_T / 2 \quad (2)$$

where l_T is the length of the scan table, while t represents time. The PM assist force, F_{PMi} , are identified from the experimentally measured force-distance relationship in [1] as

$$F_{PMi}(d_i) = ae^{-bd_i} \quad (3)$$

where $a = 1108$ N and $b = 245$ 1/m for both PM pairs.

TABLE 1. Performance specification of magnet assisted stage reported in [1].

Stage travel	0.3 m
Max. acceleration (table)	35 m/s ²
Max. speed (table)	1 m/s
Max. acceleration (AD)	10 m/s ²
Max. speed (AD)	0.15 m/s
Table size	0.36 m × 0.36 m
Identified table mass	13.8 kg

OPTIMAL CONTROL OF AADs

With the repelling PMs of the prototype stage configured as AADs, x_{PMi} can be controlled to provide maximum assist for any given x_{ref} . Representing x_{ref} , x_{PMi} , F_{LM} and F_{PMi} as $n \times 1$ vectors in discrete-time domain, with a fixed sampling interval of T_s , the linear motor force required for tracking $\mathbf{x}_{ref}$ is approximated as

$$\mathbf{F}_{LM} = m_T \ddot{\mathbf{x}}_{ref} + \mathbf{F}_{PM1} - \mathbf{F}_{PM2} \quad (4)$$

where m_T is the mass of the scan table and the dot accents represent discrete time derivatives. Note that bold face fonts are used to denote the vectorized discrete-time forms of the associated variables. Maximum assist can be achieved by generating $\mathbf{x}_{PMi}$ as the solution to a constrained optimization problem given as

$$\begin{aligned} &\text{minimize } J = \mathbf{F}_{LM}^T \mathbf{F}_{LM} \\ &\text{s.t.} \\ &\delta - \mathbf{d}_i \leq 0 \\ &|\dot{\mathbf{x}}_{PMi}| - v_{PM,max} \leq 0 \\ &|\ddot{\mathbf{x}}_{PMi}| - a_{PM,max} \leq 0 \end{aligned} \quad (5)$$

where $\delta = 0.001$ m represents the safety gap between each repelling PM pair in order to prevent collisions.

Direct Solution

A direct approach for solving the optimization problem in (5) is to treat every element of $\mathbf{x}_{PMi}$ as a variable. In which case, $2n$ variables must be

optimized simultaneously using numerical optimization algorithms. The dimension of the optimization can be halved by realizing that $\mathbf{x}_{PM1}$ and $\mathbf{x}_{PM2}$ can be solved independently. This is based on the fact that, by design, the assists from PM1 and PM2 are practically decoupled by the CV regions, during which no assist is required. However, even with a dimension of n , the direct approach can be computationally expensive because n is very large for most of practical precision scanning trajectories. With large non-convex optimization problems, the algorithms often get stuck near local optima, thus severely reducing their robustness and convergence rate. Therefore, a computationally inexpensive solution approach, which utilizes (windowed) basis splines (B-splines) to solve for $\mathbf{x}_{PMi}$, is presented and evaluated in the rest of this section.

B-spline-based Solution

B-splines, widely used for representing free-form curves [3], are adopted to reduce the number of variables in solving for the optimal $\mathbf{x}_{PMi}$. Using B-splines, $\mathbf{x}_{PMi}$ can be written as

$$\mathbf{x}_{PMi} = \mathbf{N}_p \mathbf{p}_i \quad (6)$$

where $\mathbf{p}_i$ is a $k \times 1$ control point vector and $\mathbf{N}_p$ is a $n \times k$ basis function matrix [3]. Accordingly, the k elements of $\mathbf{p}_i$ (i.e., the control points) can be used as optimization variables (where $k \ll n$), thus reducing the dimension of the optimization problem by an order of 1 or 2, typically. Moreover, $\dot{\mathbf{x}}_{PMi}$ and $\ddot{\mathbf{x}}_{PMi}$ used in the inequality constraints can be replaced by derivatives of $\mathbf{p}_i$,

$$\dot{\mathbf{p}}_i = \mathbf{D}_v \mathbf{p}_i \quad (7)$$

$$\ddot{\mathbf{p}}_i = \mathbf{D}_a \mathbf{D}_v \mathbf{p}_i \quad (8)$$

where $\mathbf{D}_v$ and $\mathbf{D}_a$ are derivative operators given as

$$\mathbf{D}_v = \frac{1}{hT} \begin{pmatrix} -1 & 1 & 0 & \cdots & 0 \\ 0 & -1 & 1 & \ddots & 0 \\ \vdots & \ddots & \ddots & \ddots & 0 \\ 0 & 0 & 0 & -1 & 1 \end{pmatrix} \in \mathbb{R}^{(k-1) \times k} \quad (9)$$

$$\mathbf{D}_a = \frac{1}{hT} \begin{pmatrix} -1 & 1 & 0 & \cdots & 0 \\ 0 & -1 & 1 & \ddots & 0 \\ \vdots & \ddots & \ddots & \ddots & 0 \\ 0 & 0 & 0 & -1 & 1 \end{pmatrix} \in \mathbb{R}^{(k-2) \times (k-1)} \quad (10)$$

with $T = (n-1)T_s$ being the duration of the scan trajectory, while h represents the spacing of the knot vector of the B-spline [3]. (Note: the knot vector is assumed to be uniformly spaced). Therefore, using (7) and (8), the constraints of (5) can be enforced indirectly using the $k \times 1$ dimensional $\mathbf{p}_i$ vectors, instead of the $n \times 1$ dimensional $\mathbf{x}_{PMi}$ vectors, thus reducing computational costs. Furthermore, with the B-spline parameterization, the time derivatives of $\mathbf{x}_{PMi}$ are continuous up to the $(m-1)$ th derivative, where m is the degree of the B-spline [3]. This means that, providing that $m \geq 3$, the smoothness of $\mathbf{x}_{PMi}$ and their derivatives is not affected by the choice of T_s . Therefore, a down-sampled T_s can be utilized to solve the optimization problem (thus reducing the computational cost). After the optimization is completed, the original T_s can then be applied to generate $\mathbf{N}_p$ in (6), thereby up-sampling the $\mathbf{x}_{PMi}$ vector back to the original resolution of $\mathbf{x}_{ref}$ before implementation on the stage.

Windowed B-spline-based Solution

Even with the adoption of B-splines, the dimension of optimization problem can still be large for most of practical precision scanning trajectories. A windowing technique can be applied, for each $\mathbf{p}_i$ vector, to sequentially solve the optimization problem in smaller chunks so as to reduce the overall computational cost of the algorithm. Let MR i represent the subset of MR regions in a scan trajectory for which the direction of acceleration of the scan table is the same. Accordingly, for each j th optimization window, a subset of $\mathbf{p}_i$ covering the j th to $(j + n_w - 1)$ th MR i regions is optimized. Note that $j = \{1, 2, \dots, n_{MR,i} - n_w + 1\}$, where $n_{MR,i}$ represents the total number of MR i regions and n_w denotes the number of MR i regions included in each window.

Evaluation of Optimization Approaches

The performance and computational cost of the three optimization approaches discussed in this section are evaluated comparatively using a simple scan trajectory (i.e., with $n_{MR,1} = 5$ and $n_{MR,2} = 6$) shown as $\mathbf{x}_{ref}$ in Figure 3. It has a scan speed of 0.8 m/s and a peak acceleration of 20 m/s² with $T_s = 0.1$ ms; $l_T = 0$ is assumed for the sake of simplicity (because l_T is a constant offset which does not make a difference in the optimization). For all the three optimization approaches, The *fmincon* function of MATLAB® is employed and run on a desktop computer with an Intel® i5 3.2 GHz quad-core processor and 12 GB memory.

With the direct method, the dimension $n = 25,001$ of the optimization problem is too large to reach a reasonable solution after several hours of computation. Therefore, $\mathbf{x}_{ref}$ is down-sampled by a factor of 50 (i.e., $T_s = 5$ ms) to solve optimization problems separately for $\mathbf{x}_{PM1}$ and $\mathbf{x}_{PM2}$. Then the resulting optimal $\mathbf{x}_{PM1}$ and $\mathbf{x}_{PM2}$ are up-sampled using cubic spline interpolation.

For the B-spline approach with and without the windowing technique, $\mathbf{x}_{ref}$ is down-sampled by a factor of 50, which is the same as for the direct approach, in order to solve the optimal $\mathbf{p}_1$ and $\mathbf{p}_2$ separately; $k = 251$ and $m = 5$ are utilized. For the windowed case, $n_w = 3$ is selected. Then the optimal $\mathbf{x}_{PMi}$ from both approaches are up-sampled using the $\mathbf{N}_p$ matrix calculated with the original $T_s = 0.1$ ms.

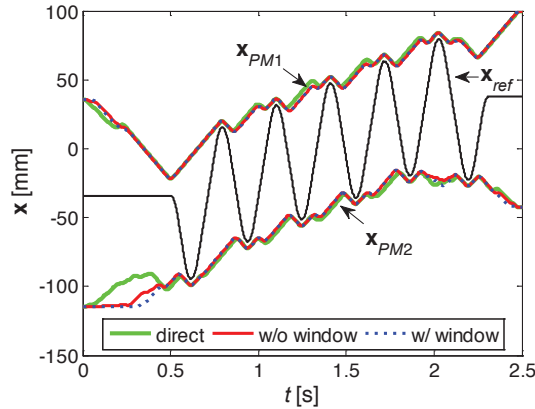


FIGURE 3. Optimal $\mathbf{x}_{PMi}$ solved by three optimization approaches (direct, B-spline without windowing, and B-spline with windowing).

TABLE 2. Comparison of computation times (T_c) and objective function values (J) for three optimization approaches (direct, B-spline without windowing, and B-spline with windowing).

Method	T_c [s]	J [N ²] (with $T_s = 5$ ms)	J [N ²] (with $T_s = 0.1$ ms)
Direct	42,270	3.54×10^6	2.08×10^8
B-spline w/o window	1,006	3.47×10^6	2.04×10^8
B-spline w/ window	257	3.47×10^6	2.04×10^8

Figure 3 shows optimal $\mathbf{x}_{PMi}$ solved using the three approaches; the $\mathbf{x}_{PMi}$ of the two B-spline-based methods are virtually coincident and are not very different from the $\mathbf{x}_{PMi}$ generated using

the direct approach. Table 2 compares the computation times, T_c , and optimal J values (before and after up-sampling) for the three optimization methods. Notice that the direct method is significantly slower than the B-spline-based approaches and yields a less optimal solution, as evident from its larger J values. On the other hand, the B-spline-based approach with windowing achieves the same J values as the one without windowing but is significantly faster.

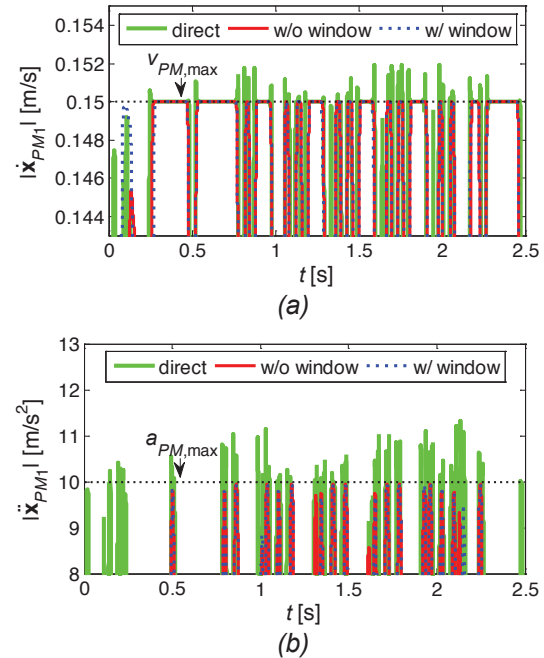


FIGURE 4. Up-sampled (a) velocity, and (b) acceleration profiles of three optimization approaches (direct, B-spline without windowing, and B-spline with windowing).

Figure 4 shows the up-sampled velocity and acceleration profiles of optimal $\mathbf{x}_{PM1}$ using the three methods. Similar plots for $\mathbf{x}_{PM2}$ have been omitted for the sake of brevity. Note that the direct approach violates the imposed constraints. The implication is that down-sampling of the direct approach to reduce computation time may lead to violation of constraints of the optimized $\mathbf{x}_{PMi}$ after up-sampling. However, it can be seen that the up-sampled velocity and acceleration profiles of the B-spline-based solutions satisfy the constraints due to the aforementioned continuity property of B-splines.

WAFER SCANNING CASE STUDIES

This section evaluates the proposed AAD against the PAD configuration of the magnet assisted stage presented in [1] using two practical case studies related to silicon wafer processing. In both case studies, the windowed B-spline approach with $n_w = 3$ is adopted for generating the optimal x_{PMi} for the AADs. For the PADs, constant x_{PMi} are obtained by minimizing the objective function J in (5) for the entire x_{ref} [1]. Note that, even though typical scan trajectories in the silicon wafer processing are composed of scan and step motions, the step motions are neglected in the presented case studies because they are beyond the scope of this paper.

Case Study 1: Scanning for Photolithography

The first case study aims to replicate the scanning pattern used in the photolithography process of silicon wafers. The following parameters are used.

- 200 mm wafer
- 25 mm x 25 mm die size
- 0.5 m/s scan speed
- 25 m/s² peak acceleration
- 25 ms settling time after MR

The scan trajectory of the wafer table are shown with gray solid lines in Figure 5 (a) describing both the scan (i.e., x-axis) and step (i.e., y-axis) motions. The dotted lines inside the solid circle line, representing the wafer boundary, are used to mark the edges of the dies. The scan tool starts from the upper left quadrant of the wafer and ends at the lower right quadrant. The beginning and ending portions of the trajectories are highlighted (with blue) for illustration purposes, while the green colored squares indicate the dies scanned by the highlighted trajectory portions and the arrows indicate the tool movement directions. B-spline parameters $k = 551$ and $m = 5$ are employed with down sampling of x_{ref} by a factor of 10 to generate the optimal x_{PMi} profiles shown in Figure 5 (b); the computation time is 1,889 s. The optimal (constant) x_{PM1} and x_{PM2} obtained for the PAD configuration scenario are 0.1274 m and -0.1274 m, respectively.

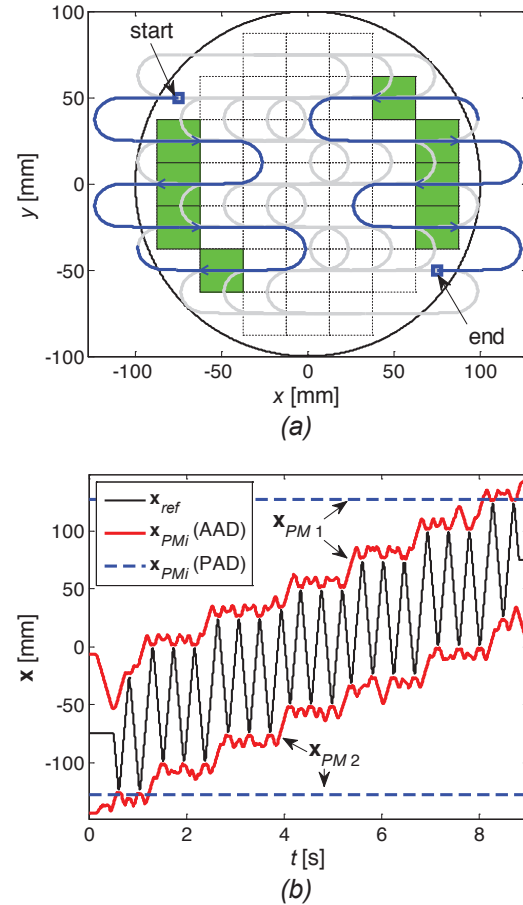


FIGURE 5. (a) Photolithography scanning trajectory, (b) Optimal x_{PMi} for AAD and PAD.

Case Study 2: Scanning for Inspection

The second case study aims at replicating the thickness measurement process of silicon wafer. The following parameters are used to generate the scanning trajectory shown in Figure 6 (a).

- 200 mm wafer
- Full length scanning with 5 mm step
- 0.25 m/s scan speed
- 10 m/s² acceleration
- 50 ms settling time after MR

As shown in Figure 6 (a), the scanning trajectory starts from the bottom and ends at the top of wafer. For the proposed AAD configuration, the optimal x_{PMi} shown in Figure 6 (b) are obtained using $k = 331$, $m = 5$ with a down sampling factor of 10; the computation time is 1,832 s. The optimal (constant) x_{PM1} and x_{PM2} for the PAD scenario are 0.1228 m and -0.1228 m, respectively.

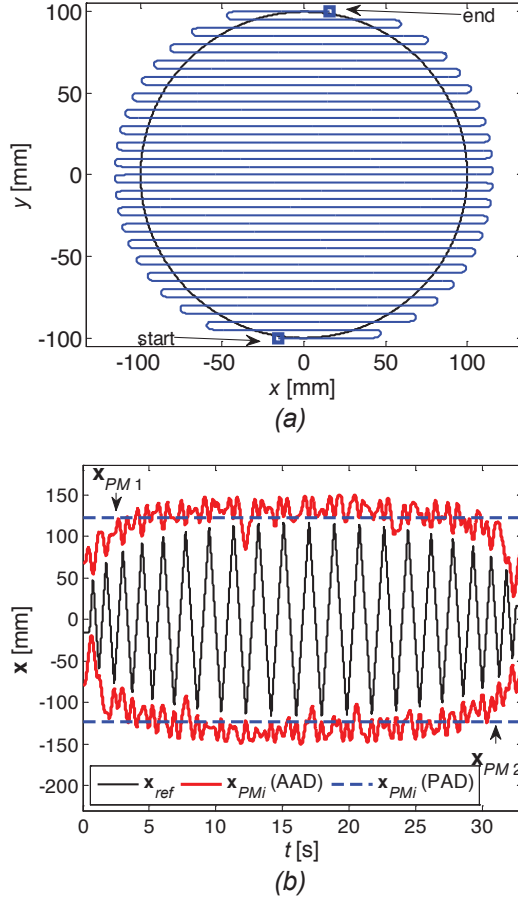


FIGURE 6. (a) Wafer inspection scanning trajectory. (b) Optimal x_{PMi} for AAD and PAD.

Experimental Results

The effectiveness of the optimally controlled AAD is experimentally validated using an in-house built magnet assisted stage prototype [1]. Three performance metrics are used to compare the effectiveness of the AAD for each case study against the PAD configuration and no PM assist scenarios; they are: (1) the RMS value of F_{LM} , measured via the motor current (i.e., $F_{LM,RMS} = (J/n)^{0.5}$), (2) the RMS value of the acceleration of the isolated base measured using an accelerometer (i.e., $a_{B,RMS}$), and (3) the RMS value of the position error during the CV regions, (i.e., $e_{CV,RMS}$) where e_{CV} is the measured position error signal inside CV regions. A 10 ms wide moving average filter is applied to the position error signal as is customary in wafer scanning operations [4]. The experimental results are summarized in Table 3 where superior performance of AAD over PAD and no PM assist cases is shown. Note that the performance gains of the PAD over the no PM assist case are marginal with trajectories of varying scanning strokes/positions.

TABLE 3. Comparison of experimental results with three stage configurations (No assist, PAD, and AAD) for two case studies.

		$F_{LM,RMS}$ [N]	$a_{B,RMS}$ [m/s ²]	$e_{CV,RMS}$ [nm]
Case Study 1	No PM	165	0.67	63.7
	PAD	158	0.64	49.3
	AAD	77	0.30	37.5
Case Study 2	No PM	48	0.18	25.1
	PAD	44	0.15	25.1
	AAD	34	0.08	13.5

CONCLUSIONS AND FUTURE WORK

An active assist device (AAD) configuration, realized with freely moving permanent magnets, has been proposed for a magnet assisted precision scanning stage. The task of optimally controlling the moving PMs to maximize assist (in order to reduce heat and vibration) is cast as a constrained optimization problem. A computationally efficient basis spline with windowing approach is presented for carrying out the optimization. The superior performance of the proposed AAD configuration in reducing heat and vibration is experimentally validated using two practical case studies related to silicon wafer processing. Future work will investigate simultaneous control of the stage's linear motor and the AADs to further reduce heat and/or vibration. Extensions of the proposed approaches to the stepping axis will also be investigated.

ACKNOWLEDGEMENT

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REFERENCES

- [1] Yoon D, Okwudire C. Magnet Assisted Stage for Vibration and Heat Reduction in Wafer Scanning. *Annals of the CIRP*. 2015; 64: 381-384.
- [2] Brown W, Ulsoy A. A maneuver based design of a passive-assist device for augmenting active joints. *Journal of Mechanisms and Robotics*. 2013; 5(3): 031003.
- [3] Piegl L, Tiller W. *The NURBS Book*. Springer. Berlin: 1997.
- [4] Butler H. Position Control in Lithographic Equipment. *IEEE Control Systems Magazine*. 2011; 31(5): 28-47.

AN OVERVIEW OF ADVANCED LIGO COMBINATION OF PASSIVE AND ACTIVE VIBRATION ISOLATION

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INTRODUCTION

To meet the very stringent seismic isolation requirements and to support the operation of its sensitive detectors, Advanced LIGO uses a unique combination of active and passive inertial isolation sub-systems. This paper explains how the isolation strategy has been defined to facilitate the design, fabrication and integration of the complex Advanced LIGO system. The article emphasizes how the combined performance of the active platforms and the passive multiple pendulums robustly supports the two detectors which are entering the first round of observation of the Advanced LIGO era.



FIGURE 1: LIGO vacuum envelope.

The two LIGO observatories located in Hanford (WA) and Livingston (LA) house 4 km long interferometers designed to detect gravitational waves. During the past five years, the new generation of instruments called Advanced LIGO has been constructed and installed in the vacuum

envelope of the two observatories [1]. FIGURE 1 shows an aerial picture of the Livingston observatory, and pictures of the vacuum envelope housing the detectors.

As of summer 2015, the first phase of commissioning of the detectors has reached completion and Advanced LIGO is entering its first observation run. This new generation of detectors should not only permit the first direct measurement of gravitational waves, predicted by Einstein's theory of general relativity, but also offer a new way to detect and analyze astrophysical events. The detectors are designed to measure the differential motion induced by gravitational waves between the test masses of the main interferometer [2]. A simplified schematic of the Advanced LIGO optical layout is shown in FIGURE 2. Two test masses are used to form 4 km long Fabry-Perot cavities in each arm of the main Michelson interferometer. Auxiliary optics are used to form mode cleaners and recycling cavities on the input and output sides of the Michelson.

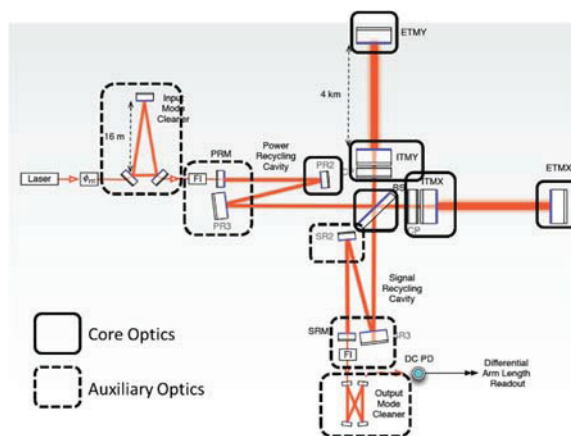


FIGURE 2: Advanced LIGO simplified optical layout (courtesy of Jeffrey Kissel).

The amplitude of the strain wave produced by astrophysical events such as binary neutron star mergers is gigantic, but such events are so distant from Earth that the amplitude of the wave is very faint when it reaches the detectors. The spectral density of the differential motion between the test masses induced by the wave is estimated to be on the order of $10^{-21} \text{ m}/\sqrt{\text{Hz}}$ around 100 Hz. To detect such a small motion, the LIGO detectors require a very high level of vibration isolation. Like most high performance isolation systems, it requires the use of multiple layers of filtering, and a subtle combination of passive and active components to obtain adequate seismic noise attenuation over a large bandwidth. The design of such distributed isolation systems is a complicated problem. Trade-offs must be made between cutoff frequencies and number of layers of isolation, while accounting for cost, schedule constraints, and robustness of operation. This paper gives an overview of strategy used for Advanced LIGO, and summarizes some of the lessons learned over this journey. The second and third sections summarize benefits and limitations of passive and active approaches respectively. The fourth section shows the approach used for Advanced LIGO, and summarizes the outcome. The last section draws conclusions.

BENEFITS AND LIMITATIONS OF PASSIVE APPROACHES

Passive approaches are often preferred for their apparent simplicity and low cost. For demanding applications, however, the natural frequencies and the number of passive layers required can drive the system complexity to levels that significantly outbalance the apparent benefits.

Assuming damping is negligible, each of the stages of an n stage isolation system, provides isolation proportional to the square of frequency above the natural frequencies of the isolator, as shown in Eq. (1). f is the frequency of vibration, f_i is the natural frequency of each individual layer of isolation, X_0 is the displacement of the ground, and X_n is the displacement of stage n .

$$\frac{X_n}{X_0} \sim \frac{\prod_{i=1}^n f_i^2}{f^{2n}} \quad (1)$$

In the presence of viscous damping, which is often necessary to reduce the resonance of the suspension modes, the filtering reduces from f^{2n} to f^n at high frequency, as shown in Eq. (2) where α_i is the damping ratio at stage i .

$$\frac{X_n}{X_0} \sim \frac{\prod_{i=1}^n 2 \alpha_i f_i}{f^n} \quad (2)$$

While active inertial solutions were proposed and studied in the early days of GW detector instrumentation [3], passive approaches were preferred for the first generation of interferometric detectors. A CAD picture of the passive stacks used in initial LIGO during the previous decade is shown in FIGURE 3. The system uses layers of stainless steel masses separated by spring dampers as shown in the picture of the stack.

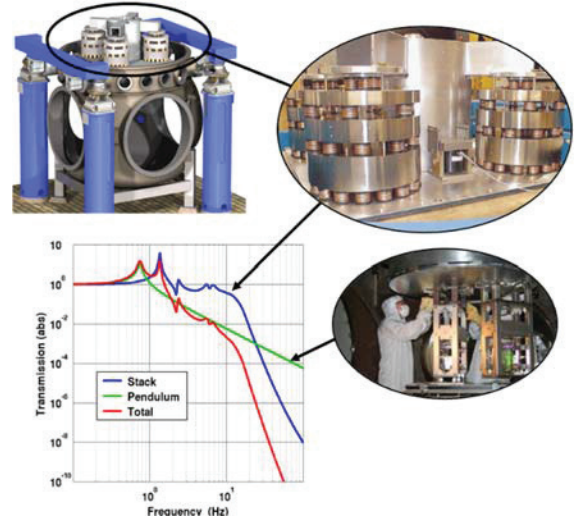


FIGURE 3: Initial LIGO passive isolation stacks.

The transmissibility of the system is shown with the blue curve in FIGURE 3. The system provides very steep isolation above the cutoff frequency which is around 10 Hz. To preserve the isolation, the spring-dampers were designed to provide only light damping, which resulted in significant resonance of the first natural mode.

To provide additional filtering at low frequencies (from 1 Hz to 10 Hz), the optics were suspended from a single pendulum system providing isolation as shown by the green curve. The overall isolation of the pendulum mounted in series with the stack is shown by the red curve. The system proved to be very effective at providing adequate isolation in the observation bandwidth (above 40 Hz), but, a low frequency active inertial isolation system had to be developed and installed at the Livingston site to improve the duty-cycle of the observatory [4].

Passive systems can be designed with lower natural frequencies to obtain further isolation, but very flexible isolators can be significantly sensitive to internal disturbances, drifts, and they can sometimes be difficult to operate.

BENEFITS AND LIMITATIONS OF ACTIVE APPROACHES

Active isolation approaches are often considered to be complex and costly. For demanding applications, they can however greatly simplify the mechanical design and speed up the construction and commissioning process.

Active vibration isolation relies on the use of inertial sensors such as seismometers, geophones and accelerometers to decouple a platform from the input disturbance. The signal of the sensor is used to measure either the disturbance in a feedforward scheme or the platform's motion in a feedback scheme. Such sensors are available in a wide variety of sizes and sensitivities, as illustrated in FIGURE 4.



FIGURE 4: Three different types of inertial sensors: seismometers, geophones and accelerometers.

Low noise instruments like broad band seismometers tend to be large and heavy while the lighter instruments such as the compact accelerometers tends to be noisier as shown in FIGURE 5.

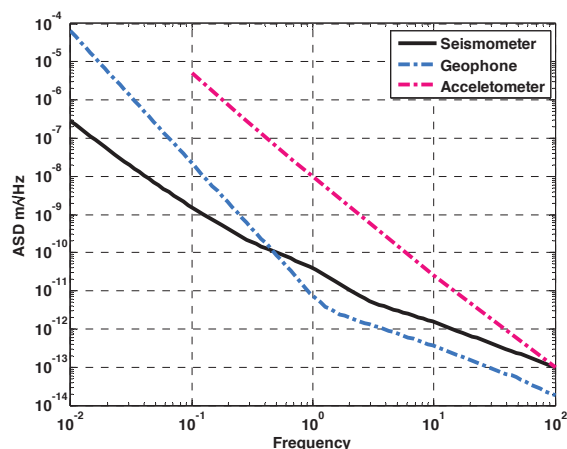


FIGURE 5: Typical amplitude spectral densities of low noise instruments.

The most sensitive commercially available instruments are the force feedback seismometers. Such sensors can be seen as a compact and robust inertial reference with effective natural frequencies of a few mHz.

Therefore an active platform servo-controlled with these instruments does not need to have particularly low natural frequencies to achieve high levels of isolation. They can be designed with relatively stiff springs, which greatly simplifies the design, the assembly process and the operation. Unlike passive approaches, active inertial feedback not only reduces the transmissibility, but also reduces the compliance, which attenuates the sensitivity to other types of disturbances.

Platforms housing those relatively heavy sensors must however be quite large, especially if the sensors must be enclosed in sealed chambers to operate in vacuum systems, as is the case in Advanced LIGO.

The performance of active feedback systems is often limited by the sensor noise and the control bandwidth. The block diagram in FIGURE 6 shows a simplified feedback control scheme, where X_0 is the input disturbance, X_1 is the platform motion to be minimized, P_s is the seismic path (transmissibility), F is the control force, and P_F is the force path (compliance). The inertial measurement of X_1 is performed with an inertial instrument which induces a sensor noise N . This signal is used to drive the control force with the compensator C .

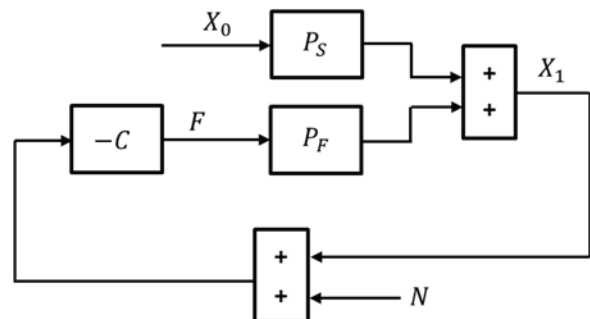


FIGURE 6: Simplified feedback block diagram.

The time averaged power spectral density of the closed loop motion can be written as shown in Eq. (3), where G is the loop gain ($G = CP_F$).

$$\langle X_1^2 \rangle = \left| \frac{P_s}{1 + G} \right|^2 \langle X_0^2 \rangle + \left| \frac{G}{1 + G} \right|^2 \langle N_x^2 \rangle \quad (3)$$

Within the control bandwidth, where the loop gain is large, the closed loop motion tends toward the expression given in Eq. (4). The first term on the right hand side is the residual motion transmission. If the loop gain is sufficiently high, the closed loop motion is limited by the sensor noise (second term).

$$\lim_{G \rightarrow \infty} \langle X_1^2 \rangle = \left| \frac{P_s}{G} \right|^2 \langle X_0^2 \rangle + \langle N^2 \rangle \quad (4)$$

The bandwidth and loop gain are often limited by the dynamics and resonances of the mechanical structure. FIGURE 7 shows examples of transfer functions for the two stages of the active platform supporting Advanced LIGO core optics [5, 6]. The rigid body modes are between 1 and 10 Hz. The lowest structural modes are around 200 Hz, which permits compensator designs with unity gain frequencies in the range of 30 Hz to 40 Hz. Controllers with higher bandwidth can be designed at the cost of less robustness.

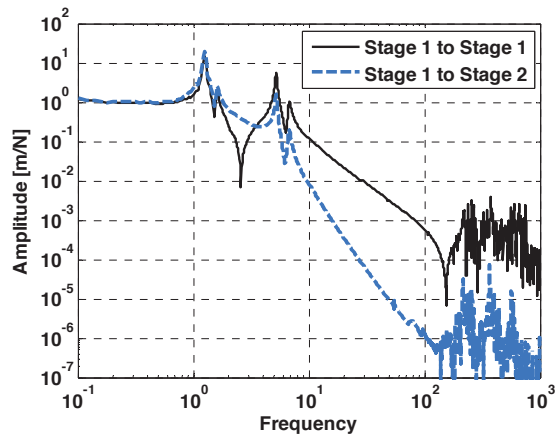


FIGURE 7: Active stage transfer functions examples.

When used as feedforward sensors, the isolation performance is limited by the coherence between the witness and the target sensors [7]. Detailed information of noise budgeting and controls optimization of active stages can be found in [8].

The next section explains how the Advanced LIGO layers of isolation have been distributed over passive and active stages, not only to meet the performance requirements, but also to facilitate the design, construction and commissioning processes.

ADVANCED LIGO COMBINATION OF PASSIVE AND ACTIVE ISOLATION

Overview

The two Advanced LIGO detectors are very complex systems which each use eleven vacuum tanks, hundreds of optical components, and servo-control loops. The system requires not only very high vibration isolation, but also high robustness and duty cycle. To achieve the objectives, the isolation strategy is based on the use of active isolation platforms instrumented

with very low noise inertial sensors, and featuring the large optical tables on which are mounted the passive suspensions holding the optics.

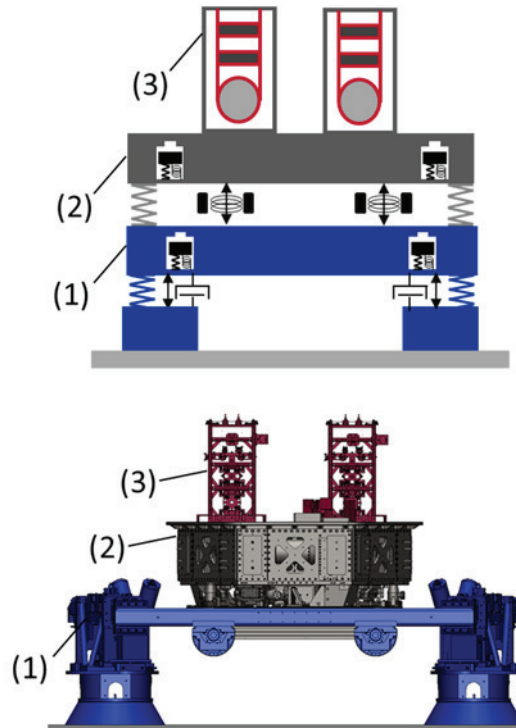


FIGURE 8: Conceptual picture (top) and CAD picture (bottom) of the layers of isolation used for the auxiliary optics of Advanced LIGO.

The schematic in FIGURE 8 illustrates the strategy used for auxiliary optics.

(1) An active stage located outside the vacuum system provides pre-isolation from approximately 0.1 Hz to 10 Hz. This hydraulic system provides a very robust base for the sub-subsequent layers of isolation.

(2) A passive-active platform features a 2 meter wide optical table on which are mounted the components of the interferometers. This system is instrumented with the large and heavy low noise inertial sensors. The system provides active isolation up to about 30 Hz, and passive isolation at higher frequencies.

(3) The optics are suspended passively using multiple pendulums. These systems provide very steep passive isolation at high frequencies. Since they are mounted on the very quiet active platforms, only light damping of the suspension modes is required.

In addition to providing seismic isolation, all of the three systems are equipped with relative sensors for positioning and alignment.

A similar approach is used for the test masses, including an additional active stage and an additional layer of passive isolation.

Active platforms

FIGURE 9 shows one of the passive-active isolation and positioning platform. It is made of a bolted aluminum structure. Cantilever springs are used to provide the vertical compliance, and flexure rods are used to provide the horizontal compliance. FIGURE 9 also zooms in on the sealed pods which encapsulate the inertial sensors to operate in ultra-high vacuum, the vacuum compatible magnetic actuators, and the position sensors used for low frequency alignment. A detailed description of the active platforms and their control scheme is given in [9].

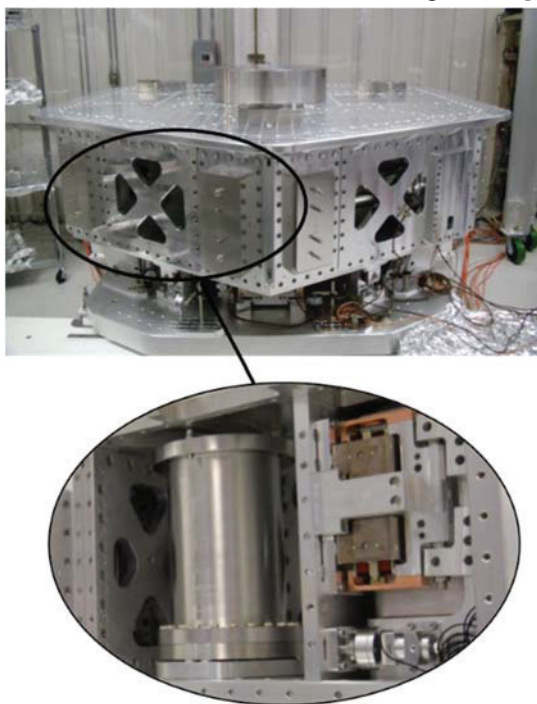


FIGURE 9: A vacuum compatible active isolator.

Passive suspensions

FIGURE 10 shows an example of a triple pendulum suspension. The surrounding welded frame supports the cantilever springs used to provide vertical isolation. Metal wires are used to suspend the stages, providing the horizontal isolation. For the core optics, silica wires are used to reduce the thermal noise within the observational frequency band [10]. The bottom stage is the mirror of the cavity. Low-noise compact and collocated actuator-sensor pairs are used to provide light relative damping of the resonances [11]. Vibration absorbers are used to damp the modes of the surrounding structure and

reduce coupling with the active platform on which it is mounted.



FIGURE 10: A triple pendulum suspension.

Integration, performance and operation

The balanced passive-active approach developed for Advanced LIGO greatly facilitated the construction and commissioning process of the very complex detectors [12].

FIGURE 11 shows an example of isolation results obtained with the active platforms. The black curve shows the ground motion. The red and blue curves show the motion of two of the active platforms, which provide very similar isolation. The motion is near or below requirements at all frequencies above 0.5 Hz. The low frequency performance is currently limited by tilt-horizontal coupling in the inertial sensors [13].

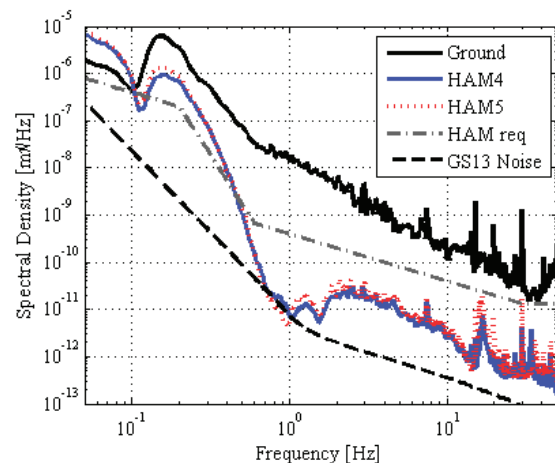


FIGURE 11: Active platform performance

In FIGURE 12, the measurements of the six degrees of freedom of the active platform are combined to estimate the longitudinal motion at the input point of the suspension. In the red curve, this motion is combined with a model of the isolation provided by the triple suspension. As shown in this example, only light damping of the resonance is needed. The combined active and passive isolation permits bringing the optic motion under $10^{-16} \text{ m}/\sqrt{\text{Hz}}$ at 10 Hz.

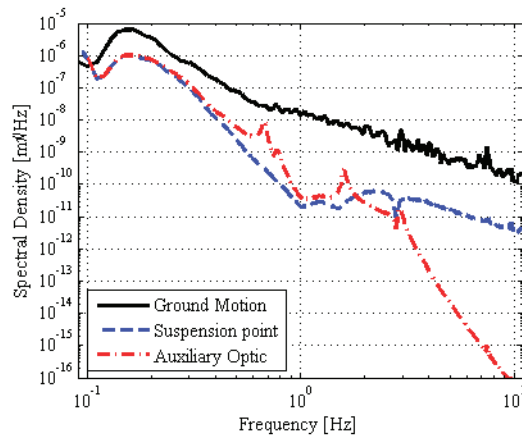


FIGURE 12: Combined active-passive isolation.

CONCLUSION

The Advanced LIGO vibration isolation strategy is based on very-low-noise stand-alone active platforms, on which are mounted multi-stage passive suspensions. The integration and commissioning process of Advanced LIGO proved that approach to be very effective. During the project construction, eleven active pre-isolators, ten active isolation platforms, and tens of passive suspensions have been constructed and integrated at each of the two LIGO observatories, on schedule and on budget. The combination of active and passive isolation systems permit robust operation of the two detectors as they enter the first observation run of the Advanced LIGO era.

ACKNOWLEDGEMENTS

LIGO was constructed by the California Institute of Technology and Massachusetts Institute of Technology with funding from the National Science Foundation, and operates under cooperative agreement PHY-0757058. Advanced LIGO was built under award PHY-0823459. The material presented in this paper is the fruit of the outstanding work of the LIGO Suspension and Seismic isolation teams. This paper carries LIGO Document Number P1500165. FIGURE 5 was previously used in [14] published in the Journal of Sound and Vibration, and is re-used with permission from Elsevier.

REFERENCES

- [1] Aasi, J., et al., *Advanced LIGO*. Classical and Quantum gravity, 2015. **32**(7): p. 074001.
- [2] Abbott, B., et al., *LIGO: the laser interferometer gravitational-wave observatory*. Reports on Progress in Physics, 2009. **72**(7): p. 076901.

- [3] Robertson, N., et al., *Passive and active seismic isolation for gravitational radiation detectors and other instruments*. Journal of Physics E: Scientific Instruments, 1982. **15**(10): p. 1101.

- [4] Wen, S., *Improved Seismic Isolation For The Laser Interferometer Gravitational Wave Observatory With Hydraulic External Pre-isolator System*. 2009, Ph.D Thesis, Louisiana State University.

- [5] Matichard, F., et al., *Advanced LIGO two-stage twelve-axis vibration isolation and positioning platform. Part 1: Design and production overview*. Precision Engineering, 2015. **40**: p. 273-286.

- [6] Matichard, F., et al., *Advanced LIGO two-stage twelve-axis vibration isolation and positioning platform. Part 2: Experimental investigation and tests results*. Precision engineering, 2015. **40**: p. 287-297.

- [7] DeRosa, R., et al., *Global feed-forward vibration isolation in a km scale interferometer*. Classical and Quantum Gravity, 2012. **29**(21): p. 215008.

- [8] DeRosa, R.T., *Performance of Active Vibration Isolation in the Advanced LIGO Detectors*. 2014, University of Rochester.

- [9] Matichard, F., et al., *Seismic isolation of Advanced LIGO: Review of strategy, instrumentation and performance*. Classical and Quantum Gravity, 2015. **32**(18): p. 185003.

- [10] Aston, S., et al., *Update on quadruple suspension design for advanced LIGO*. Classical and Quantum Gravity, 2012. **29**(23): p. 235004.

- [11] Strain, K. and B. Shapiro, *Damping and local control of mirror suspensions for laser interferometric gravitational wave detectors*. Review of Scientific Instruments, 2012. **83**(4): p. 044501.

- [12] Staley, A., et al., *Achieving resonance in the Advanced LIGO gravitational-wave interferometer*. Classical and Quantum Gravity, 2014. **31**(24): p. 245010.

- [13] Matichard, F. and M. Evans, *Review: Tilt-Free Low-Noise Seismometry*. Bulletin of the Seismological Society of America, 2015. **105**(2A): p. 497-510.

- [14] Collette, C. and F. Matichard, *Sensor fusion methods for high performance active vibration isolation systems*. Journal of sound and vibration, 2015. **342**: p. 1-21.

Flexure Design of Active Microarchitected Materials That Achieve Programmable Properties via Control

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ABSTRACT

The aim of this paper is to introduce the flexure design for a new kind of microarchitected material, called an active microarchitected (AM) material, that can be programmed to achieve desired combinations of actively tunable properties via principles of swarm control. Such materials consist of a lattice of repeating unit cells that behave as tiny self-contained mechatronic systems complete with actuators, sensors, flexures bearings, and the circuitry necessary to actively control the overall lattice's bulk mechanical properties (e.g., Young's modulus, shear modulus, Poisson's ratio, and damping coefficients). The proposed material design is able to exhibit the programmed properties in a repeatable and useful way because the relative motions of its structural constituent elements are guided and actuated by precision flexure bearings and non-contact actuators and sensors that minimize internal friction and hysteresis. This new material would enable a number of advanced technologies including: (i) aircraft wings that can selectively alter their stiffness properties at desired locations according to changing wind conditions for increasing maneuverability and fuel efficiency, (ii) advanced vibration damping structures onto which sensitive instruments can be attached (e.g., laser range finders), (iii) barriers that attenuate, amplify, or channel sound in desired directions, and (iv) materials that store memories pertaining to where, when, and by how much they have been deformed or impacted to assess if and where potential damage has occurred. Approaches for fabricating AM materials is discussed.

ACTIVE MICROARCHITECTURED MATERIAL

Traditional microarchitected materials [1] consist of large lattices of tiny structured cells that passively exhibit a desired set of bulk properties based solely on the engineered topology and material properties of each cell's constituent microstructural elements. In contrast, AM materials, like the example shown in Fig. 1A,

consist of cells that behave as tiny self-contained mechatronic systems that can be actively controlled using internal actuators and sensors to achieve desired bulk properties.

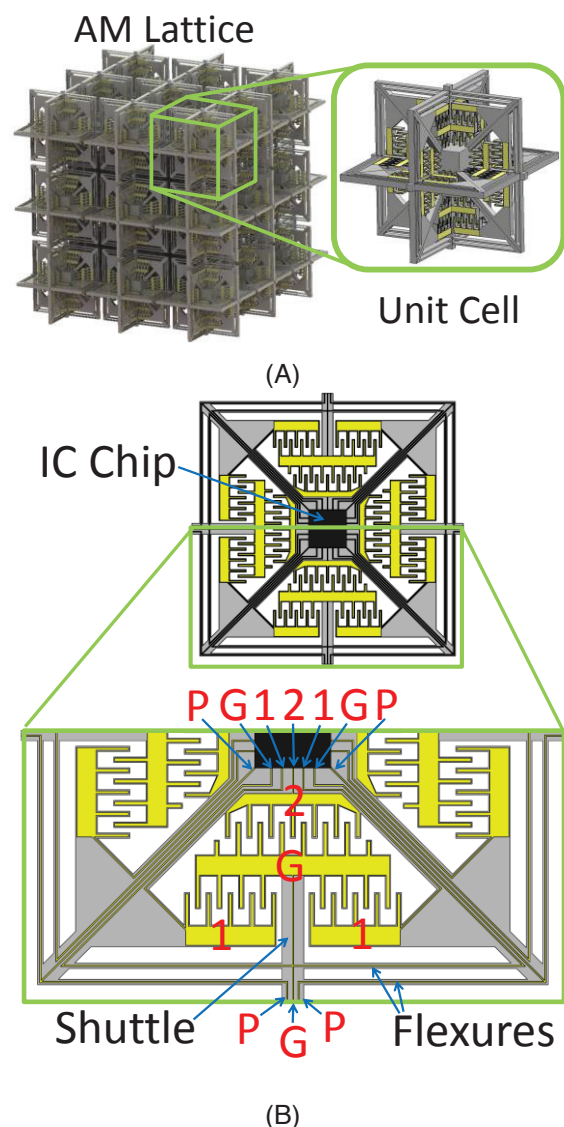


FIGURE 1. AM material example (A) with a 2D cross-section of its unit cell showing the cell's electrical wiring (B)

A 2D cross-section of the unit cell from Fig. 1A is shown in Fig. 1B to clarify how the design could work. A programmed integrated circuit (IC) chip would control the actuation forces imparted on the shuttles within each cell. Electrostatic comb drives would actuate and sense [2] the position of these shuttles, which are constrained by flexure bearings to only allow translations along their axes and are connected to the shuttles of their neighboring cells. The IC chips could be externally powered and electrically grounded according to the lines labeled *P* and *G* respectively, and they could drive the shuttle's electrically grounded combs by supplying voltages to the combs labeled 1 and 2. Note that yellow areas are electrically conductive while grey areas are not. In this way, the entire material's lattice could be programmed to achieve actively tunable mechanical properties. Thus, AM materials can achieve extreme mechanical properties that could not be achieved by naturally occurring materials, composites, or other passive microarchitected designs because AM materials utilize an active power source that can significantly amplify their response to mechanical loads.

The approach used to control the mechanical properties of such AM material lattices utilizes principles of swarm control. Such control is inspired by natural systems (e.g., colonies of ants or schools of fish) that exhibit emergent behaviors or patterns as a result of the collective behavior of thousands of individual entities. In the case of AM materials, the IC chips within the individual cells of the lattice can be programmed with simple commands for responding to shuttle displacements caused by the movements of neighboring cells to create bulk lattice properties that emerge from the individual behaviors programs within each individual cell.

PRECISION FLEXURE DESIGN

One of the enabling features that allow AM materials to achieve their programmed properties in a repeatable way is the precision flexure bearings that guide the shuttles of the lattice's unit cells. Although the unit cell design of Fig. 1 demonstrates the concept of the proposed AM materials of this paper, the design's flexure topology is problematic. If the main bodies of any two neighboring cells are held fixed with respect to each other, their shared shuttle is free to vibrate back and forth along its axis because the design is under-constrained [3]. This is a problem because each

cell's IC chip has no way of knowing if the bulk lattice is being loaded or if its shuttles are merely vibrating. A sophisticated flexure design is necessary for guiding the desired shuttle degrees of freedom (DOFs) while also fully constraining all of the internal rigid bodies such that the shuttle motions will always correspond with the bulk loading scenario (i.e., the flexures must not permit under-constraint).

The flexure topology must also possess the required planes of symmetry necessary to ensure that the resulting passive bulk properties will be as close to isotropic as possible. For 3D designs that consist of a single repeating unit cell that fits within a space-filling polyhedron (e.g., the unit cell of Fig. 1A that fits within a cube), 9 unique plans of symmetry are necessary to achieve cubic property directionality [4], which is as close to isotropic as such designs can be. These 9 planes of symmetry appear as four planes spaced 45° from each other when seen from the perspective of the three orthogonal cross-section views of the unit cell (e.g., the cross-section view shown in Fig. 1B).

Suppose flexures were added to the 3D unit-cell design of Fig. 1A such that one of its three orthogonal cross-sections looked like Fig. 2A instead of Fig. 1B. This new unit-cell design would not only possess the optimal planes of symmetry (shown as red lines in Fig. 2A) similar to the design of Fig. 1, it would also possess rigid bodies that are not under-constrained. Thus, unlike the under-constrained design of Fig. 1 with a controller that could become unstable due to the vibrations along the directions of motions that are not constrained sufficiently by flexures, the design of Fig. 2 would avoid this issue entirely. A 2D cross-section of the 3D lattice is shown in Fig. 2B. If the central rigid bodies of any two neighboring unit cells are held fixed with respect to each other, it is clear from the expanded view shown in Fig. 2B that the shuttle shared by both cells would also be fully constrained and would possess no DOFs. The unit cells, however, would still be free to move toward or away from each other as desired given the flexure topology as long as the cells are not held fixed with respect to each other. Thus, the design is properly constrained to enable the stable control of programmable bulk lattice properties. Note also that the 3D design of Fig. 2, shown as a 2D cross-section, could be simplified and made to

exhibit less over-constraint while still satisfying the desired design requirements previously discussed (e.g., optimal planes of symmetry and a flexure topology that is not under-constrained) if the original flexures labeled in Fig. 1B were removed. If, however, the design of Fig. 2 were not a cross-section of a 3D design, but rather a planar 2D version of that design, the original flexures labeled in Fig. 1B would be necessary to maintain in order to achieve proper constraint.

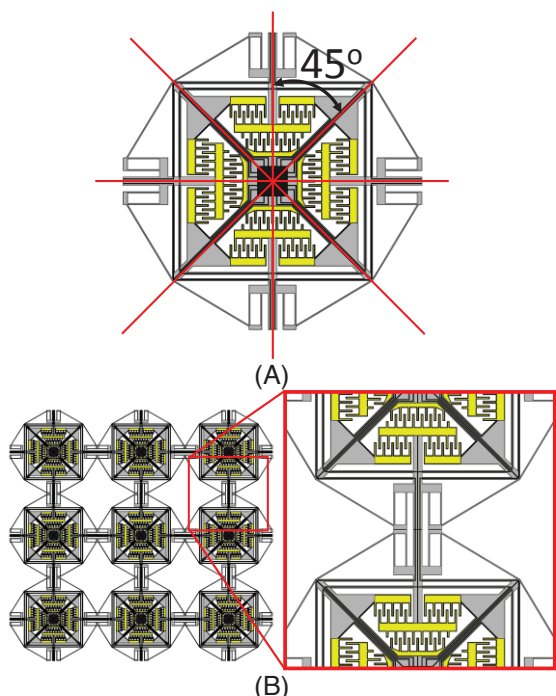
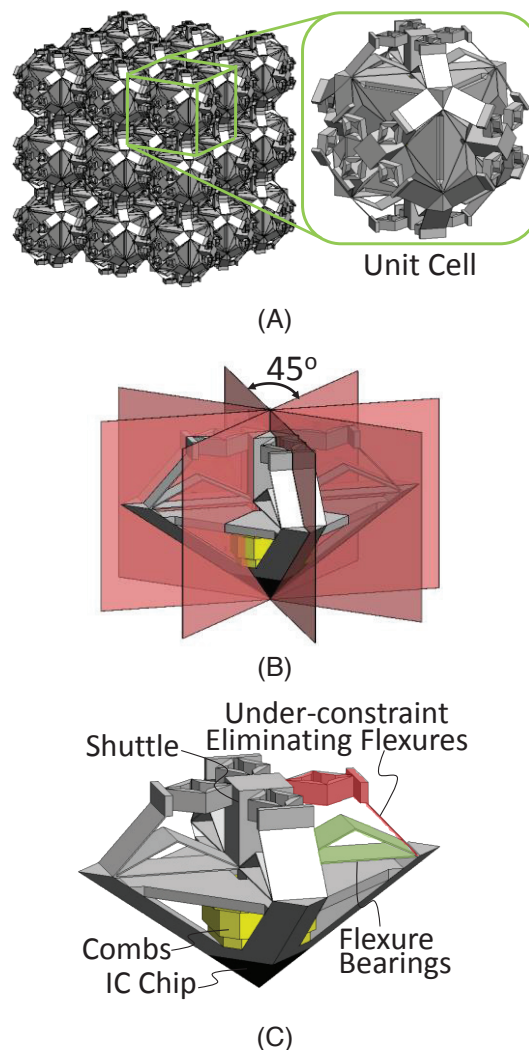


FIGURE 2. 2D cross-section of an AM material unit cell with flexures that eliminate under-constraint (A); lattice showing how these unit cells join (B).

A more sophisticated 3D design is shown in Fig. 3A. This design could be made to possess more electrostatic combs per available volume and could thus access a larger range of achievable programmable properties. One of the design's sectors is shown in Figs. 3B and 3C. This sector is repeated six times within each cube-shaped unit cell in the lattice. Because this sector possesses the four planes of symmetry spaced 45° as shown in Fig. 3B, the design achieves the desired cubic material property directionality discussed previously. The design's flexure topology is also not under-constrained. One set of flexure bearings that guide the desired translation of the designs' shuttles are colored green and labeled in Fig. 3C. These flexures are analogous to the flexures from the designs in

Figs. 1 and 2 labeled "Flexures" in Fig. 1B (i.e., they perform similar tasks). Another set of flexures that eliminate under-constraint are colored red and labeled in Fig. 3C. These flexures are analogous to the flexures that were added to the design of Fig. 1 to create the design of Fig. 2. How the flexure topology of the design of Fig. 3A works is best visualized by considering the two sectors joined together as shown in Fig. 3D. If the central rigid bodies of these two sectors are held fixed with respect to each other, the shuttle shared by both sectors would also be fully constrained and would possess no DOFs. The sectors would be free, however, to move toward or away from each other as desired given the flexure topology as long as the sectors are not held fixed with respect to each other. Thus, the design is properly constrained to enable the stable control of programmable bulk lattice properties. A cross-section of one of the sectors (Fig. 3E) shows the



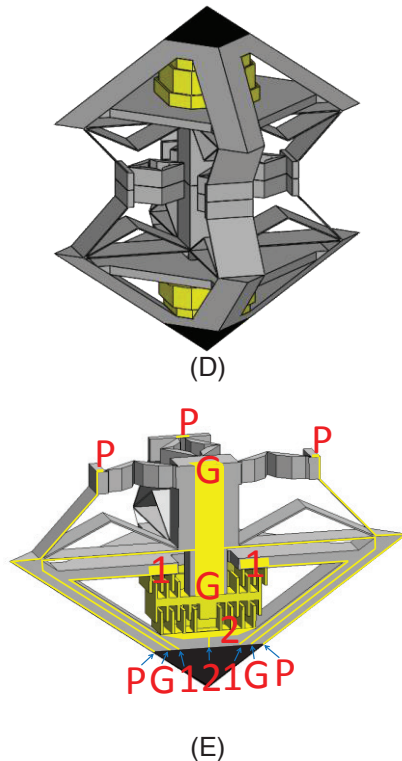


FIGURE 3. Another 3D AM material lattice with a more optimal flexure topology (A); its unit cell sector showing the desired planes of symmetry (B); the cell's functional parts (C); two joined sectors showing how the flexures within the lattice would eliminate under-constraint (D); the unit cell's cross-section showing electrical wiring (E).

electrostatic comb region as well as the wiring that enables the IC chip to control the location of its shuttle. Again, the cell's IC chip could be externally powered and electrically grounded according to the lines labeled *P* and *G* respectively. These programmed IC chips could also drive the shuttle's electrically grounded combs by supplying voltages to the combs labeled 1 and 2 in Fig. 3E. Note again that yellow areas are electrically conductive while grey areas are not. Note also that the wiring that connects the IC chips, actuators, and sensors is redundantly configured throughout the entire microarchitected material so that large numbers of lines can afford to be faulty while still allowing most of the IC chips and combs to receive their intended voltages such that the desired bulk properties can still be achieved. One of the strengths of such AM material designs is that even if many unit cells are damaged or fail to operate as intended, the overall bulk material response will be largely

unaffected due to massive redundancy in the system.

The flexure topologies of the AM material concepts provided in this paper were designed using the Freedom and Constraint Topologies (FACT) synthesis approach [5]. The advanced under-constraint eliminator flexures were designed using an extension of this theory [6].

POTENTIAL FABRICATION APPROACH

Although micro-additive fabrication approaches are soon to realize the mechanical and electrical infrastructure of 3D AM materials that consist of non-conductive and conductive constituent materials, additive fabrication will not likely be able to fabricate their IC chips any time soon. Thus, if such materials are to become a reality in the near future, their IC chips will need to be independently fabricated using their current processes, and then they can be inserted inside the material's unit cells while they are being printed. Although the initial cost to develop IC chips using existing processes is high, producing IC chips in bulk quantities is relatively inexpensive. And, since IC chips can currently be made at appropriately small sizes ($\sim 1\text{mm}^2$) and are becoming smaller and smaller with every advancing year, existing micro-additive approaches (e.g., projection micro-stereo lithography [7]) could be used in conjunction with existing IC chip fabrication processes to fabricate AM materials at their intended size in the not-too-distant future. The bottom portion of each unit cell would likely be fabricated first using additive fabrication, and then prefabricated IC chips would be mechanically inserted and wire bonded inside each cell in parallel using a robotic manipulator (Fig. 4A). Then the remaining top portion of each cell would be fabricated using additive fabrication (Fig. 4B), and this process would be repeated layer by layer.

CONCLUSIONS

This paper introduces the concept of AM materials that utilize swarm control to achieve programmable and actively tunable mechanical properties. Specifically, the paper addresses the topological design of the precision flexure bearings within a variety of AM material concepts that are actuated and sensed by non-contact electrostatic combs. The topologies of these designs were synthesized using FACT such that they (i) possess the appropriate planes of symmetry necessary to achieve desired

material property directionality, and (ii) are not under-constrained. Potential fabrication approaches are discussed for making these designs a reality at their intended scale.

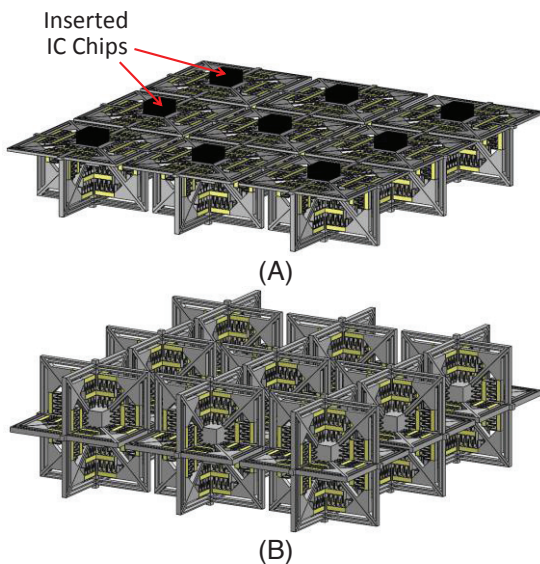


FIGURE 4. IC chips inserted into half-printed cells (A); the rest of the cells are then printed on top (B) and the process continues layer by layer.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] Gibson, L.J., Ashby, M.F., 1997, *Cellular Solids: Structure and Properties—Second Edition*, Cambridge University Press, Cambridge, United Kingdom.
- [2] Dong, J., and Ferreira, P. M., 2008, "Simultaneous Actuation and Displacement Sensing for Electrostatic Drives," *Journal of Micromechanics and Microengineering*, 18(3): pp. 35011-35020.
- [3] Blanding DL. *Exact constraint: machine design using kinematic principles*. New York, NY: ASME Press; 1999.
- [4] Cowin, S. C., and Mehrabadi, M. M., 1995, "Anisotropic Symmetries of Linear Elasticity," *Appl. Mech. Rev.*, 48(5), pp. 247-285.
- [5] Hopkins, J.B., 2013, "Designing Hybrid Flexure Systems and Elements Using

Freedom and Constraint Topologies," *Mechanical Sciences*, 4: pp. 319-331.

- [6] Hopkins, J.B., Panas, R.M., "A Family of Flexures That Eliminate Underconstraint in Nested Large-Stroke Flexure Systems," *Proc. of the 13th International Conference of the European Society for Precision Engineering & Nanotechnology*, Berlin, Germany, May 2013.
- [7] Beluze, L., Bertsch, A., and Renaud, P., 1999, "Microstereolithography: A New Process to Build Complex 3D Objects," *Proc. SPIE Conference on Design, Test, and Microfabrication of MEMS and MOEMS*, Paris, France, 3680, pp. 808-817.

Hybrid Additive and Microfabrication of an Advanced Micromirror Array

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ABSTRACT

We have developed a hybrid conventional and additive manufacturing (AM) technique for the fabrication of a high fill-factor (99%) micro-mirror array design (10,000 1mm^2 mirrors). The micromirrors are driven with three degrees of freedom (DOFs)—tip, tilt, and piston—over large ranges ($\pm 10^\circ$ rotation and $> \pm 30\mu\text{m}$ translation) at high speeds ($\sim 40\text{kHz}$ small stepping rate), all with continuous closed-loop control. The capabilities of this new mirror array will extend the performance of a variety of high-impact technologies including: (i) optical switches, (ii) confocal microscopes, (iii) autostereoscopic display and image capture, (iv) microprojectors, (v) high speed focusable LIDAR or other imaging, (vi) micro-additive fabrication approaches that utilize principles of light steering (e.g., two photon polymerization or optical tweezers), and (vii) high-powered laser steering systems. The initial fabrication effort is focused on the generation of an independently controllable seven-mirror prototype. The fabrication process performance and limitations are described.

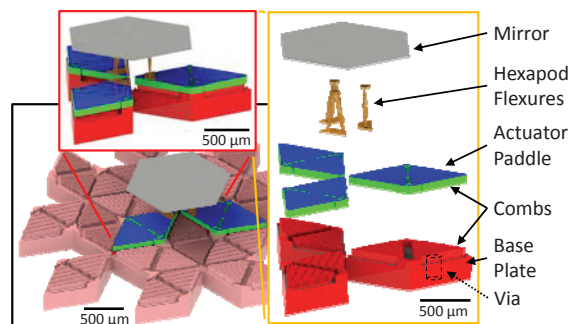


FIGURE 1: Micromirror array design.

STRUCTURE

A seven mirror array is shown in Fig. 1 to illustrate the scale and coR. mponents of the micromirror. Only the middle element is shown fully assembled. The micromirror array is composed of hexagonal unit cells, each of which contains three bipolar electrostatic comb drive actuator paddles, three decoupling flexure linkages and a hexagonal mirror. The actuators are the green/blue paddles which are anchored at two corners to the red actuation plate below and are free to rotate around a single axis.

PROCESS OVERVIEW

The mirrors are assembled from i) a microfabricated actuation plate and ii) micro-stereolithographically generated hexapod flexures, which are printed onto the underside of iii) microfabricated silicon hexagonal mirrors. The finished structure is shown in Fig. 2. The process limitations were characterized for both the microfabrication and the AM side. An initial pick-and-place operation was used to generate a single assembled cell. An improved parallel assembly process is being developed for handling all 7 mirrors to generate the 7 element array.

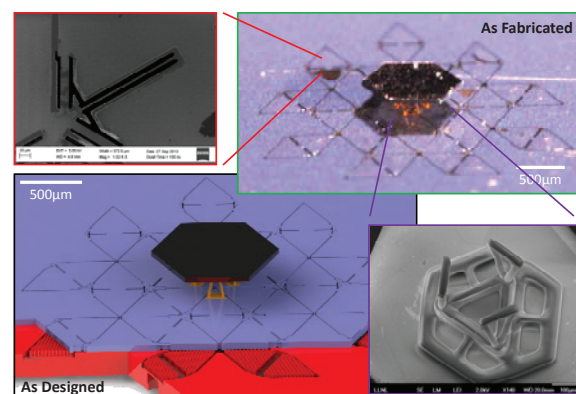


FIGURE 2: Fabricated vs. designed structure.

¹These are both first authors as their contributions to this paper are equal

MICROFABRICATION

The actuation plate is made of two microfabricated single crystal silicon wafers, shown in Fig. 1 as the actuator paddle and the base plate. These two plates are aligned and fused together.

MAIN PLATES

The ground paddles comprise the flexible moving rhomboid stage above each set of combs. These are all fabricated as part of a double-SOI single crystalline wafer. The top surface of the ground plate must be flat and smooth as it will form one side of the bond to the actuation plate. The combs on the ground plate are etched, producing the comb patterns within each paddle as shown in Fig. 3.

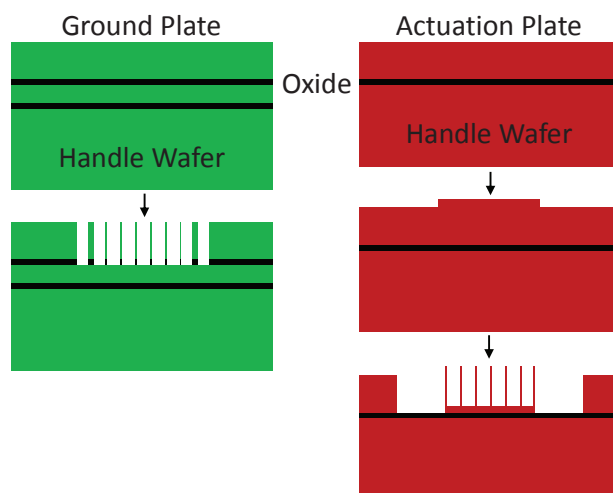


FIGURE 3: Ground plate etching.

The fabrication process next focuses on the actuation plate, which is the non-moving side with the isolated islands of combs. The actuation plate is started as an SOI wafer where the top layer is highly doped (conductive) Si, and the bottom layer is undoped (non-conductive) Si intended to act as the structural layer. Three conflicting requirements are found at the surface of this plate. 1) The top surface of the actuation plate must be flat and smooth as it will form one side of the bond to the ground plate. 2) The combs should slightly overlap one another to provide a smooth and uniform change in capacitance through the equilibrium position. 3) The combs should be made of single crystal Si to maximize their strength. The selected growth of a uniform layer of epitaxial silicon over the comb islands was found to meet all three of

these requirements. The surface is patterned and the combs are etched to produce electrically isolated comb 'islands'.

BONDING AND BACK ELECTRONICS

The two wafers are now flipped and a BCB bond was first used to attach the wafers. Later efforts have focused on wafer fusion bonding for the conductive contact and higher alignment accuracy. This step determines the comb alignment, and so must be done to high accuracy. The present target is $<2\mu\text{m}$.

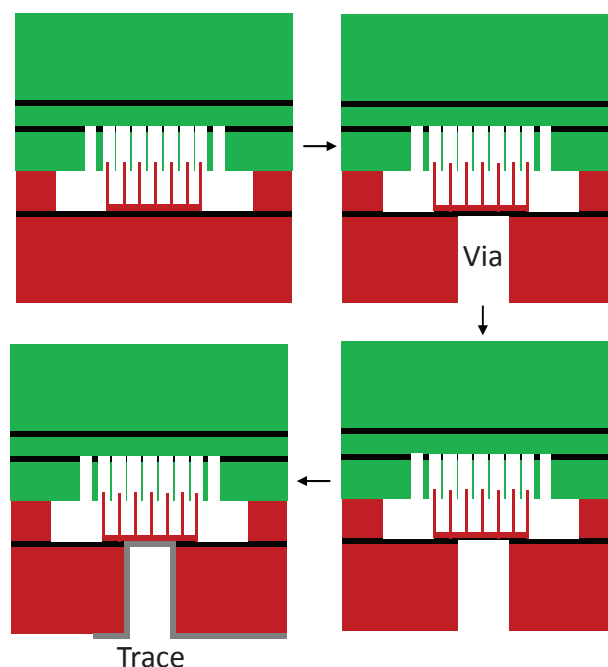


FIGURE 4: Bonding and thinning.

The bonded stack is next etched from the back to form vias through the actuation plate handle wafer up to the oxide behind each comb island. The oxide is removed to provide direct electrical contact to the comb islands. Traces are then deposited down the side of the vias to provide connectivity through the wafer and out the back.

STACK THINNING AND RELEASE

The bonded stack is thinned down to the first oxide layer on the ground plate side, at which point the oxide is used to help form a two-step etch that frees the actuator paddles as well as forming the in-plane blades. The bonding and thinning steps are shown in Fig. 5.

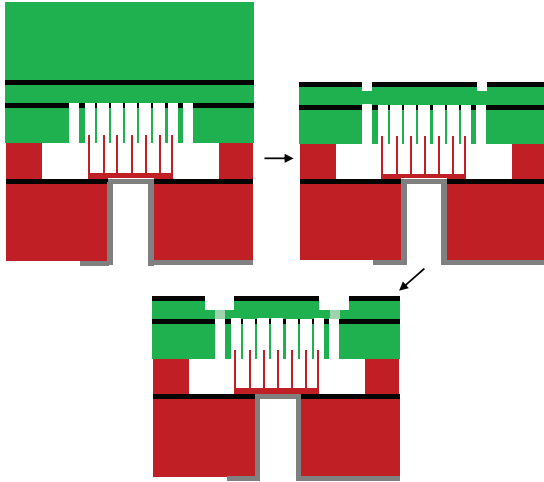


FIGURE 5: Stack thinning and release.

This forms the full structure, ready for attachment to the additively manufactured parts.

The resulting combs in the process are shown in Fig. 6 below and shows the desired comb array.

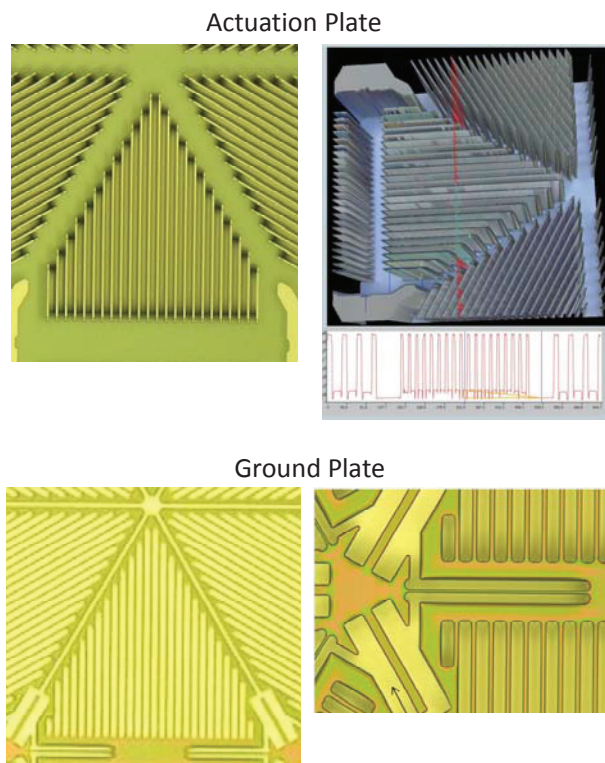


FIGURE 6: Formed combs on both sides of the actuator.

ADDITIVE MANUFACTURING

The hexapod flexures, shown in Fig. 1, are generated by Projection micro-Stereolithography

(PμSL) [1], carried out by exposing sequential layers of HDDA onto the back of the silicon mirrors. The performance limits of the PμSL system with both HDDA and PEGDA was studied to determine the minimum leg stiffness as shown in Fig. 7. HDDA was used due to the similar small scale performance but with reduced moisture sensitivity.

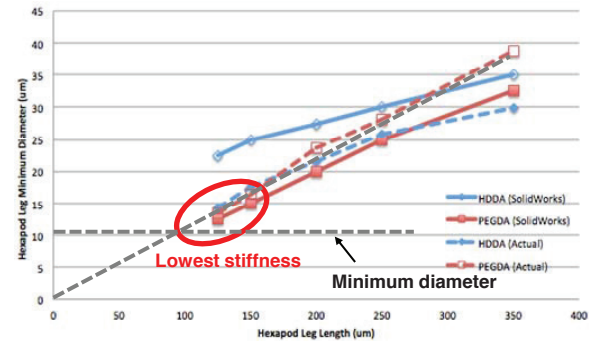


FIGURE 7: Dimensional Limits for PμSL.

MIRRORS

The mirrors are fabricated in a batch process from an SOI wafer, where the device layer is cut into hexagonal patterns, gold coated, then released to form free mirrors that can be placed into the array.

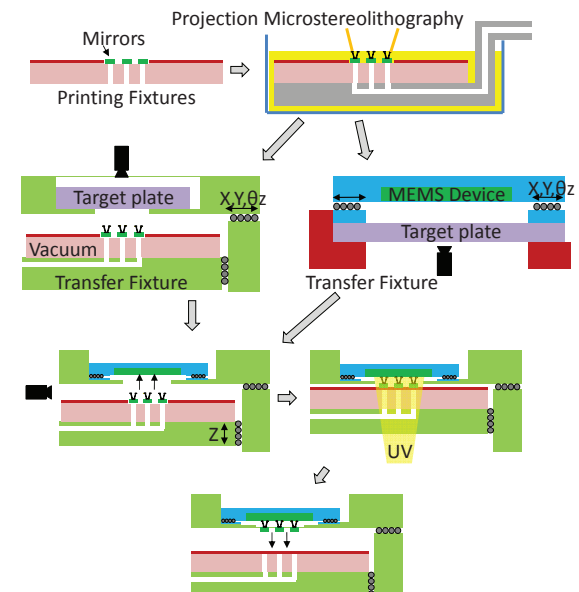


FIGURE 8: Formed combs on both sides of the actuator.

HANDLING PROCESS

The mirrors are handled using two distinct fixtures. The first 'printing' fixture holds the seven mirrors in the correct pattern for the AM

printing process. This fixture ensures the hexapod legs are printed down in a uniform pattern, and provides a means to handle the array as a single element for transfer to the final MEMS structure. The printing fixture is passed into a 'transfer' fixture, as shown in Fig. 8.

The transfer fixture is used to align the MEMS device to the hexapod+mirror array. This is done via a five step process. The first step is aligning the hexapod array to a fiducial target plate. This is done via a monocular microscope looking directly down through the glass target plate, shown in Fig. 9.

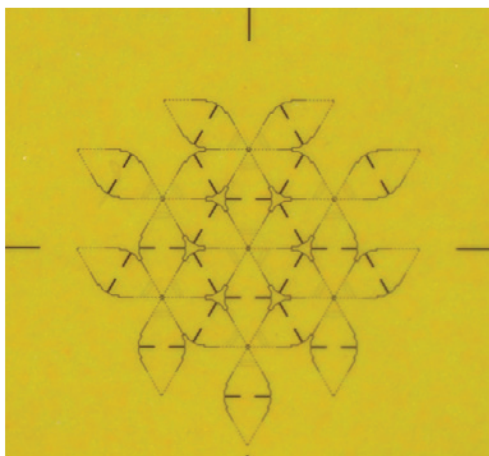


FIGURE 9: Fiducial marks for transfer fixture alignment.

The glass fiducial plate serves as a target to align the printing fixture to the overall frame of the transfer fixture. This process is repeated with the MEMS device aligned to an identical glass fiducial plate. The result is both the hexapod legs and the MEMS structure are aligned to the transfer fixture coordinate frame. At this point, the fiducial plates are removed, the MEMS and AM structures are touched together and the UV epoxy on the hexapod legs is cured. The mirrors are released via a positive pressure through the printing plate.

PRINTING FIXTURE

This fixture must simultaneously meet several requirements. It must be able to controllably and rigidly hold and release the mirrors. The fixture must provide a frame to help the mirror placement approximate that of a hexagonal array. The fixture must protect the coated side of the mirrors from the AM bath. The fixture must not alter the AM process chemistry. The surface of the fixture must have a large wetting angle

with the polymer bath in the AM process, or else the thin layers of HDDA would not be able to form over the mirrors. The fixture must be resistant to the AM bath cleaning alcohols. These requirements led to the design of a multi-material printing fixture as shown in Fig. 9 and 10.

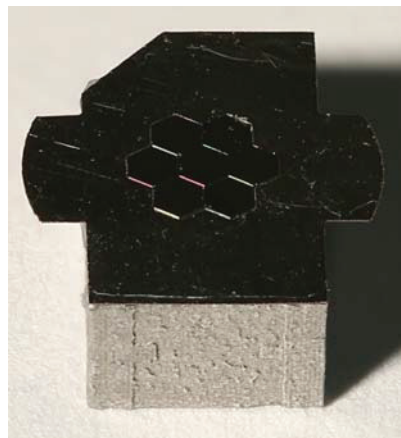


FIGURE 9: Printing fixture with all seven mirrors inserted.

The printing fixture is composed of an outer metal frame, with a silastic polymer cast within. The top surface of the fixture is covered with a steel plate that has a laser cut frame pattern to hold the mirrors in an array pattern.

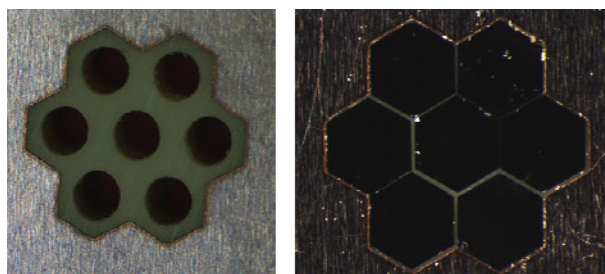


FIGURE 10: Top down view of printing fixture with vacuum holes and polymer visible on the left, then with the mirrors placed on the right.

This design meets all the required criteria. It has vacuum holes behind each mirror so that positive or negative pressure can be applied to the mirrors to hold or release them on command. The steel frame and top plate provide a reference frame that can hold the mirrors in an array pattern. The mirrors are placed face down against the internal polymer, and the vacuum suction holds them tightly against the polymer. This provides a robust seal against the AM bath fluids contaminating the front side. The polymer surface was found to

have a low wetting angle, but the use of the steel plate on the surface mitigates that issue. Finally, the materials of the fixture are resistant to alcohols.

The printing fixture is placed on a submersible arm with a vacuum passage in it, which provides suction to hold the mirrors in place while they are in the P μ SL bath. The hexapod structures are sequentially printed onto the back of the mirrors in a controlled array, resulting in the structure as seen in Fig. 11.

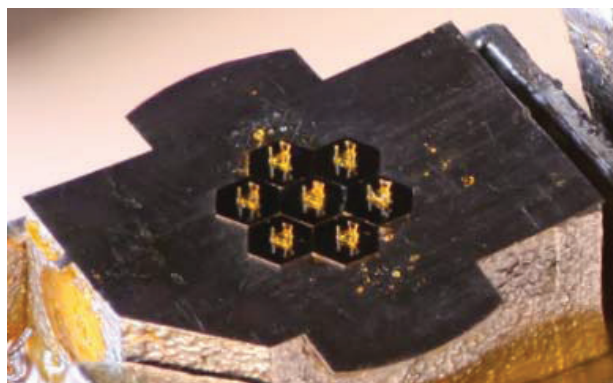


FIGURE 11: Seven element array of printed hexapods on the seven silicon mirrors.

The printing fixture is then passed to the transfer fixture for alignment to the MEMS device.

TRANSFER FIXTURE

The transfer fixture is composed of a Z-axis bearing mounted on an X-Y stage. This all lies within an optical θ_z mount which provides the full degrees of freedom needed for the alignment of the parallel structures to one another. The transfer fixture has a vacuum port to maintain the vacuum applied to the mirrors, until positive pressure is required to release the mirrors.

The fixture is shown in Fig. 12 with the printing fixture installed in the center post. This post is able to slide vertically to drive the hexapod legs up against the MEMS device. Before the final transfer is done, the hexapod legs are brought up against the glass fiducial plate, placed in a kinematic frame on the top of the structure. The X-Y stage and the θ_z stage are used to align the hexapod legs to the fiducial to within a few μm .

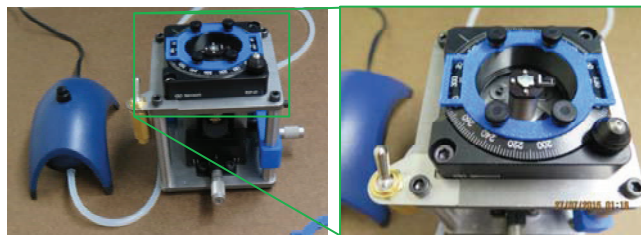


FIGURE 12: Mirrors/hexapods transfer fixture.

The same process is repeated with the MEMS device on a second fixture, shown in Fig. 13. This second transfer fixture has the same degrees of freedom but a different stage structure, so the MEMS chip can be firmly anchored in place.



FIGURE 13: MEMS transfer fixture.

This second fixture aligns the MEMS chip to the fiducial plate, at which point the MEMS chip can be transferred via kinematic mounts to the primary fixture and pressed against the hexapod legs for final transfer.

RESULTS

The assembly process is presently ongoing, with early tests of AM structure transfer to substrates underway. The early results are promising for the ability to do repeatable and fine motion control.

Difficulties have been observed with ensuring a vacuum seal between the mirrors and the polymer internal structure of the printing fixtures. When this fails, the vacuum system can be flooded with HDDA, resulting in the mirrors floating off the fixture. Allowing HDDA behind the mirrors also increases the release force required to pull the seven mirror array off the printing fixture during the final transfer to the MEMS device. This force must be on the few-mN scale to avoid damaging the MEMS

structure paddles, so careful control of this force is required. Ongoing efforts are focused on decreasing detritus on the fixture surfaces, and using the controllable back-pressure to drive a counteracting load on the mirrors. This offers the potential to negate the stiction forces if properly tuned.

CONCLUSIONS

A new method is described for the fabrication of a high-speed large-range micromirror array. This fabrication process draws upon both conventional microfabrication and emerging micro-AM processes to generate a 3D microscale optomechanical system.

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REFERENCES

1. Ha, Y. M, et. al., "Mass production of 3-D microstructures using projection microstereolithography," *Journal of Mechanical Science and Technology*, v 22, pp 514-521, 2008.

SUB-5NM DUAL-FIELD OVERLAY CONTROL IN JET AND FLASH IMPRINT LITHOGRAPHY USING TOPOLOGY OPTIMIZATION

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1. INTRODUCTION

This paper describes a method to realize sub-5nm overlay control simultaneously in two lithography fields, using the technique of *topology optimization* for the template design. The dual field overlay method has been developed for Jet and Flash Imprint Lithography (J-FIL), which is a specific form of Imprint Lithography [1]. J-FIL is described in detail in [2] and is being considered as a low-cost alternative to more expensive photolithography extension techniques such as self-aligned double/quadruple patterning (SADP/SAQP). Canon's J-FIL based manufacturing tools for NAND Flash and DRAM manufacturing are described in a recent paper [3].

Precise overlay has been demonstrated for J-FIL using the magnification and scale control system [4] (MSCS), with recent demonstration of sub-5nm overlay [3]. This level of overlay, however, has been shown only over a single lithography field (26mm x 33mm).

This paper is focused on multi-field overlay control, specifically dual-field overlay, to exploit an important advantage of imprint lithography relative to photolithography – the ability to increase patterning area without significantly impacting the complexity of the patterning tool. While such large area imprinting has been demonstrated widely in the literature (see for e.g. [2]), these demonstrations have not included nano-scale overlay. Multi-field imprinting has the ability to dramatically impact the cost of memory fabrication due to increased throughput. Sub-10nm overlay using a dual-field template and MSCS was explored before by Yin [5], but it was not possible to achieve sub-5nm dual-field overlay using the uniformly solid template in Yin's work because of mechanical coupling between the two fields. This is because error correction in one field induces undesirable errors in the other field. A novel way to reduce the mechanical coupling is by machining holes at carefully chosen locations in the template outside the active printing area. The machining can be performed by leveraging an existing process for fabrication of J-FIL templates which involves precisely machining out a circular

region that is approximately 1 mm thick in a 6025 photomask blank [4].

2. SINGLE AND DUAL-FIELD OVERLAY CONTROL USING THE MSCS – A BRIEF BACKGROUND

For single field overlay control, eight alignment correctibles are usually compensated for – translation in X and Y, rotation (θ), magnification correction in X and Y, orthogonality (γ), and trapezoidal correction in X and Y (see Fig. 1). The first step in a field-by-field alignment scheme is to compare alignment marks placed at the four corners of each field to corresponding marks on the previous patterned layer. The eight alignment values obtained are then linearly transformed into the eight alignment correctibles. Alignment is performed in two steps. First, low viscosity resist is dispensed onto the wafer and the template brought in contact with the dispensed resist. Next, the rigid body displacements (X and Y translation and θ) are corrected by the wafer stage. The five scale/shape errors (X,Y magnification, gamma, and X,Y trapezoid) are then corrected by applying precisely controlled forces at the periphery of the template using the MSCS. Additional details on the MSCS can be found in the work of Cherala et al. [3].

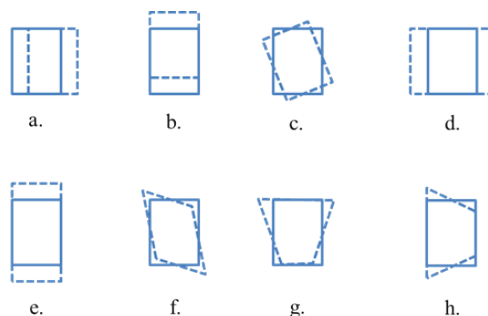


FIGURE 1 (a) Translation X, (b) Translation Y, (c) Rotation (θ), (d) X Magnification, (e) Y Magnification, (f) Orthogonality (γ), (g) X Trapezoid, (h) Y Trapezoid.

As mentioned previously, the idea of dual-field overlay has been proposed by Yin. In Yin's scheme, the pair of fields is imprinted as shown

in Fig. 2. With this configuration close to two-thirds of the wafer fields can be imprinted two at a time, giving an overall throughput gain of about 1.5x.



FIGURE 2. A pair of dual-imprinted fields, shown among an array of fields on a 300mm wafer.

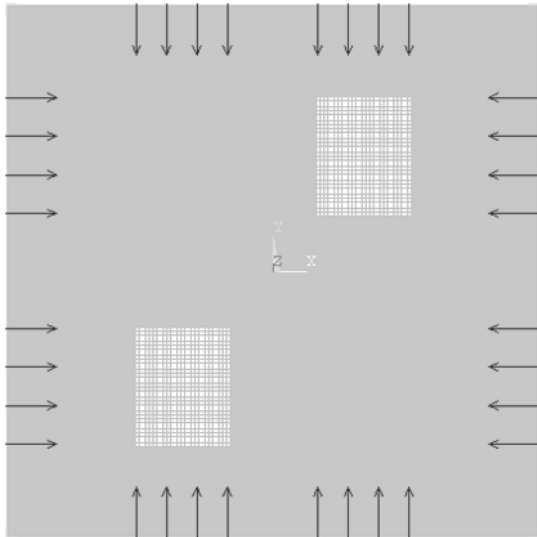


FIGURE 3. Representation of a template based on a 6025 mask blank for dual-field imprint. The array of white grid points defines a field. The two fields are in Yin's dual field configuration. Forces are applied at the base of the arrows at the edges of the template. The length of the arrows represents sample force magnitudes.

The overlay control of Yin's dual-field scheme is similar to the single field scheme of Cherala. Correction forces are applied at the periphery of the template to bring about the required overlay correction. Errors are measured using alignment marks located at the corners of each field. The difference is that the template now has two fields instead of one (see Fig. 3).

Peripheral correction forces can be calculated as follows –

1. Template deformation at grid points is described by the following equation:

$$C * \{F_{template}\} = \{\delta\}$$

$\{F_{template}\}$ is the boundary 32x1 force vector, $\{\delta\}$ is the template deformation vector at the grid points and C is the template compliance matrix.

2. Three force components of $\{F_{template}\}$ must be used for maintaining static equilibrium of the template. These three will be linearly dependent on the remaining force components. $\{F_{template}\}$ can be expressed in terms of the independent force combinations as follows:

$$\{F_{template}\}_{noFx1} = \{FN_{template}\}_{noFx(noF-3)} \alpha_{noFx1}$$

noF is the total number of template correction forces, and, α_{noFx1} is the degree of activation of the basis set of statically balanced forces $\{FN_{template}\}_{noFx(noF-3)}$.

3. The original deformation equation can now be rewritten as:

$$\{C * \{FN_{template}\}\} * \alpha = \{\delta\}$$

$C * \{FN_{template}\}$ can be evaluated by solving the general plate deformation equation for unit activations of the basis force combinations using finite element analysis ($C * \{FN_{template}\}$ will be referred to as CFN from here on).

4. Finally, $\{F_{template}\}$ can be found by inverting CFN . Usually, CFN is not a square matrix since the number of grid points is much larger than the number of actuators. Additionally, there are a variety of constraints on the magnitude and direction of forces that can be applied onto the template. Therefore, only an approximate solution can be obtained. For these force-constrained, non-square cases, correction forces are found by formulating an optimization problem for minimizing the maximum x-y overlay error.

3. DESIGN OF TEMPLATE FOR IMPROVED DUAL-FIELD OVERLAY USING TOPOLOGY OPTIMIZATION

As mentioned before, Yin's dual-field scheme is not capable of sub-5nm dual-field overlay control (see Table 1) since it uses a uniformly solid template. This problem with Yin's scheme can be circumvented by machining holes in the template, outside the active printing area, which result in an overall reduction in the inter-field mechanical coupling. Calculating the number, location, shape and size of the holes is done using topology optimization.

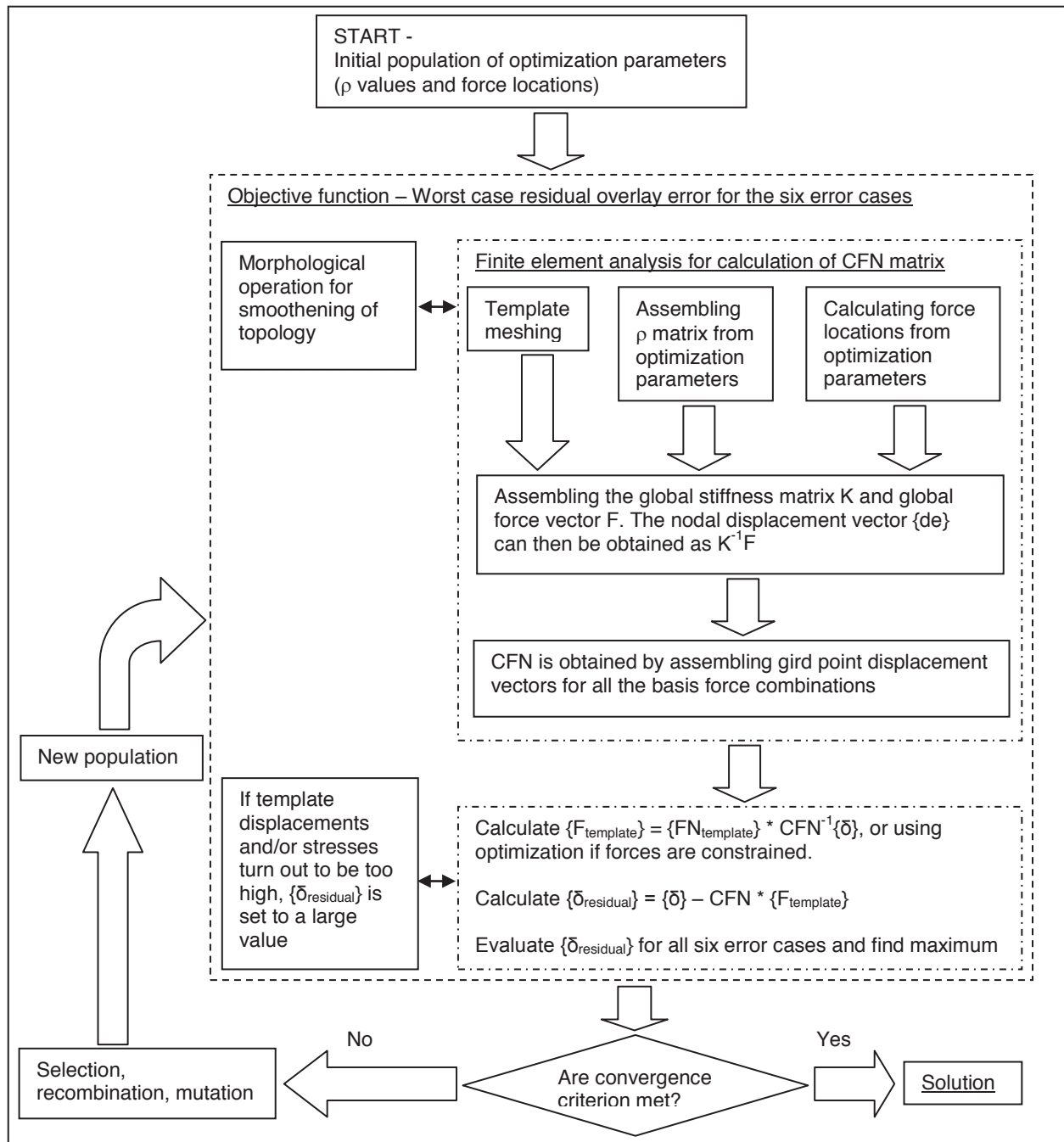


FIGURE 4. Schematic of the template topology and force location optimization.

3.1 Algorithm for topology and force-location optimization of JFIL templates

Fig. 4 shows a schematic of the optimization algorithm. The algorithm is built around a plane stress finite element analysis of the template. For the finite element analysis, the template needs to be meshed to calculate displacements associated with boundary forces. For the topology optimization, a density factor “ ρ ” is additionally associated with each meshed

element. ρ can either be one or zero (to avoid computational problems a small number is used instead of zero), corresponding to the presence or absence of material at that element. This way any topology of the template can be represented in terms of an array of ρ values. Four geometrical parameters locate the forces around the template. Given the meshed state, the ρ matrix and the force locations and magnitudes, a finite element solution can be obtained. CFN can

then be obtained by finding finite element solutions for all the basis force combinations.

Only wafer grid and 1st order intra-field errors are considered in the optimization algorithm. Residual overlay error is calculated for the following error cases: 1ppm X Mag, 1ppm Y Mag, 10nm X Translation, 10nm Y Translation, 1urad Gamma and 1urad Theta, each of which is applied only on Field 1. These cases consist of 6 linear error types commonly corrected in optical and imprint lithography systems (along with commonly encountered error magnitudes). Any general dual-field linear error case can be decomposed in terms of the above six cases. Thus, good performance for these cases would generally ensure good performance for any mixed linear error case.

The output of the objective function is the maximum of the six residual error values obtained above. Genetic algorithm (GA) is used to minimize this output. The reason for choosing GA over other more efficient gradient based techniques (Bendsoe and Sigmund [6]) is that GA requires only first order function calculations. The objective function in our case is neither continuous nor differentiable. Gradient based optimizers cannot work with such an objective function. Matlab's GA toolbox is used for the actual implementation. The optimization starts with an initial population of optimization parameters and the objective function is repeatedly evaluated (for all members of the current population) until some kind of convergence criterion is met. Members within the current population are selected according to performance, recombined with other members and mutated, generally resulting in better performing populations as generations progress.

4. PHOTOELASTICITY BASED EXPERIMENTS

The phenomenon of photoelasticity can be used to validate the optimization procedure for specific error cases. Transparent polymeric materials, such as polycarbonate, exhibit birefringence when subjected to stress. The magnitude of birefringence is proportional to the magnitude of shear stress in the material. Thus, different shear stresses in the two fields in a dual-field template optimized for decoupling gamma error would show up as a difference in the brightness of the two fields.

The experimental setup (see Fig. 6) consists of an array of 16 individually controllable force fingers, arranged around the periphery of a square photoelastic template (PSM-1 photoelastic material by VPG). The photoelastic

template is machined at to optimally decouple gamma (orthogonality) error between the two outlined fields. An analyzer and a polarized white light source are placed above and below the photoelastic plate to visualize the stressed regions.



FIGURE 6. Image of a section of the experimental setup showing the machined template, force fingers, and the two fields.

5. RESULTS

5.1 Results – Simulations Showing Alignment Correction Capability

Table 1 shows the performance of the optimized template in comparison to a uniformly solid template. Residual overlay error for the optimized template is well below 5nm, and significantly better than the uniformly solid template for most cases. Figure 7 shows the optimized template.

It should be noted that the simulation results of Table 1 assume tensile forces can be applied on the template. It is difficult to actually pull as well as push on the template in the Magnification and Scale Control System (MSCS). Tensile forces can still be applied by deliberately introducing a fixed error in the template (during fabrication), which is equal and opposite to the error introduced by a uniform compression of the template from all sides. Thus, by starting off with this same uniform compression of the template, the fixed initial error is cancelled out. Tensile forces can then be “applied” by reducing the compressive force in a specific actuator.

TABLE 1. Performance comparison between Yin's dual-field template and the topology (and force location) optimized template (all residual overlay errors are in nm)

Initial Error Template	1ppm X Mag	1ppm Y Mag	10nm X Trans	10nm Y Trans	1urad Gamma	1urad Theta
Yin's (dual field)†	4*	6.4	0.9	1.1	7.9	5.5
Topology and force location optimized (dual field)†	1.9	2.5	2.7	2.3	3.2	2.9
*Residual overlay error, in nm, after applying correction †Limiting forces = $\pm 50\text{N}$ Tensile forces are allowed on the template						

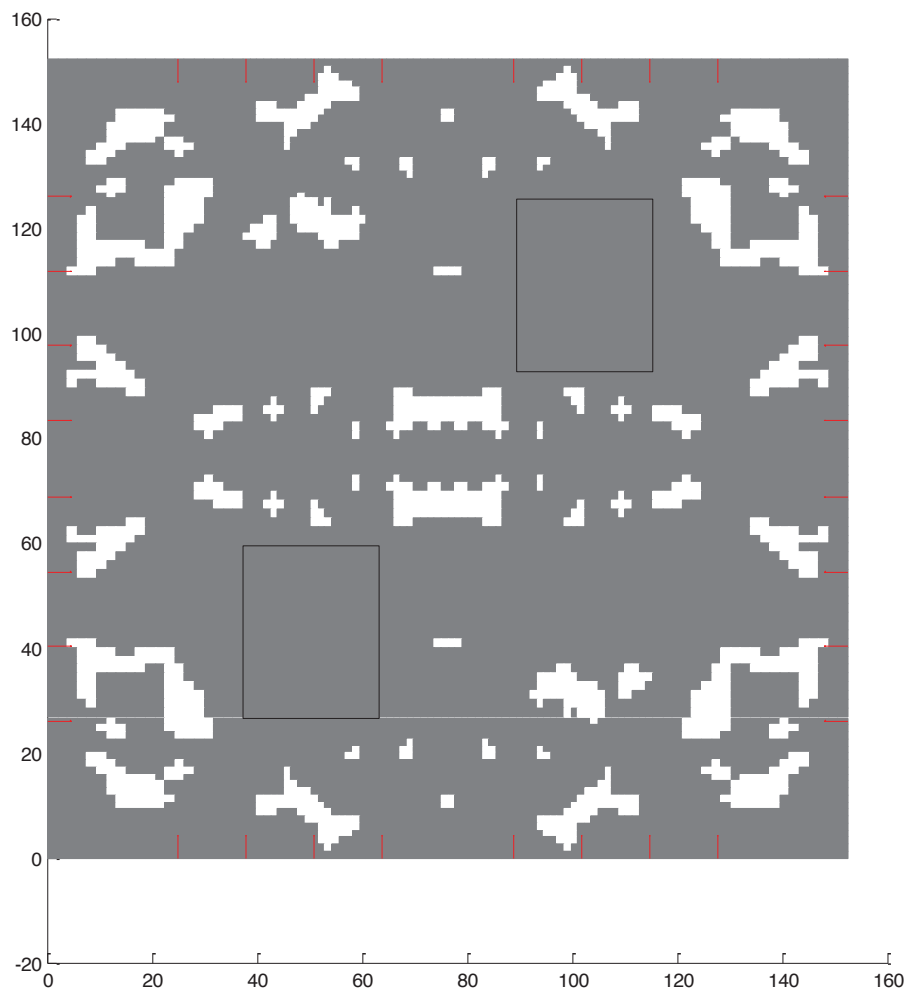


FIGURE 7. Topology and force location optimized template. Material is present only in the grey regions. The black outlined regions are the two fields. The red arrows along the periphery are the force application locations.

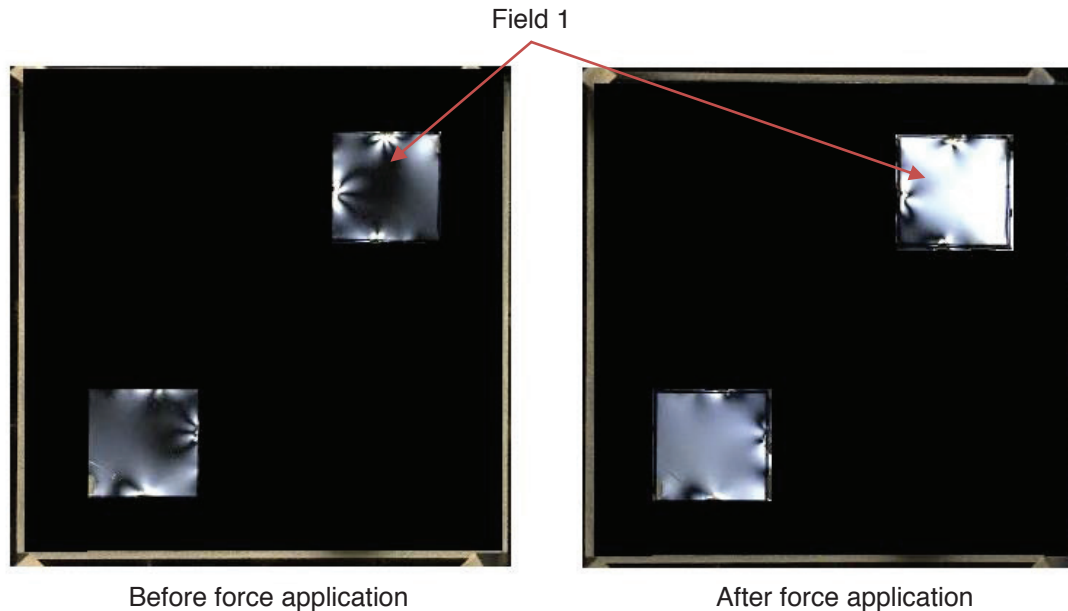


FIGURE 8. Photoelastic template before and after force application, seen under a polariscope (template regions outside of areas of interest have been blacked out). The presence of bright regions before force application is due to machining defects.

5.2 Results - Photoelastic Experiments to Validate Simulations Used in Earlier Section

In this section, experiments were performed to validate the simulation used in Section 5.1. These validation experiments cannot demonstrate the dual-field overlay algorithms demonstrated earlier because the hardware available for this experiment only included 16 actuators (rather than the 32 actuators needed for dual field overlay).

Fig. 6 shows a template which has been machined to optimally decouple γ distortion between Fields 1 and 2. It should be noted that the template used here is different from the template of Fig. 7 and has been optimized for the specific case of γ distortion, along with the additional constraints of no tensile forces and 16 actuators and is again only being used to validate the simulation. The template is optimized for γ distortion as it shows up well in photoelastic measurements.

Simulations show that the $\mu+3\sigma$ value of the distortion in Field 1 should be about two times that in Field 2. Fig. 8 shows the experimental result which is in agreement with the simulation. It can be seen that Field 1 experiences a greater change in brightness than Field 2, which implies that the optimized template selectively induces γ distortion in Field 1 without affecting Field 2 significantly.

6. SUMMARY

It has been demonstrated that, using the technique of topology optimization, dual-field overlay can be achieved in J-FIL with sub-5nm residual distortions. Photoelasticity experiments have been used to validate the genetic algorithm based simulations used to design the template and MSCS force locations.

7. ACKNOWLEDGEMENTS

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8. REFERENCES

- [1] S. Chou, "Nanoimprint lithography". JVST B. vol. 14, pp. 4129-4133. 1996.
- [2] S.V. Sreenivasan, "Nano-Scale Manufacturing Enabled by Imprint Lithography". MRS Bulletin, Special Issue on Nanostructured Materials in Information Storage. Vol. 33, pp. 854-863. Sept. 2008.
- [3] H. Takeishi, S.V. Sreenivasan, "Nanoimprint System Development and Status for High Volume Semiconductor Manufacturing". Proc. of SPIE Advanced Lithography. Feb. 2015.
- [4] A. Cherala *et al.*, "Nanoscale Magnification and Shape Control System for Precision Overlay in Jet and Flash Imprint Lithography". Mechatronics, IEEE/ASME Transactions on. Vol. no.99. pp.1,11. 2014.
- [5] B. A. Yin, "Dual Field Nano Precision Overlay". Master's Thesis, The University of Texas at Austin. 2010.
- [6] M.P. Bendsoe, O. Sigmund, "Topology Optimization – Theory, Methods and Applications". Springer-Verlag

CARBON NANOTUBE GROWTH FORCE DETECTION ON MULTI-AXIS MEMS SENSOR WITH INTEGRATED MICROHEATER

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ABSTRACT

This paper presents the fabrication and calibration of a multi-axis microelectromechanical (MEMS) force sensor used to measure the growth force of carbon nanotubes (CNTs) grown by chemical vapor deposition (CVD). A tungsten microheater located on the center stage of the force sensor is used to provide localized thermal energy to achieve CVD growth. Force resolution of the sensor is designed to detect the force exerted by CNT during growth which is estimated to be 16 pN. Sensor calibration methods are presented for the polysilicon piezoresistive force sensors and microheater.

INTRODUCTION

CNTs have been an exciting material for over two decades due to their exceptional mechanical and electrical properties. However, their

integration into consumer products has been limited by one factor – manufacturing. This stems from a limited understanding of the growth mechanism [1].

To examine the growth mechanism of the CNTs from a mechanical perspective, a multi-axis force sensor with an integrated microheater was developed. This sensor is used to measure the force generated by CNTs during growth. In the CVD growth measurement process shown in Figure 1, a CNT initiates growth from a catalyst particle on the force sensor, Fig. 1.b). Carbon atoms join together in a tubular structure to form the CNT which grows from the catalyst towards a separate substrate. Once the CNT makes contact with the substrate, the resultant force is measured by the force sensor, Fig. 1.c). A previous experiment estimated the growth force of CNTs to be approximately 16 pN by studying

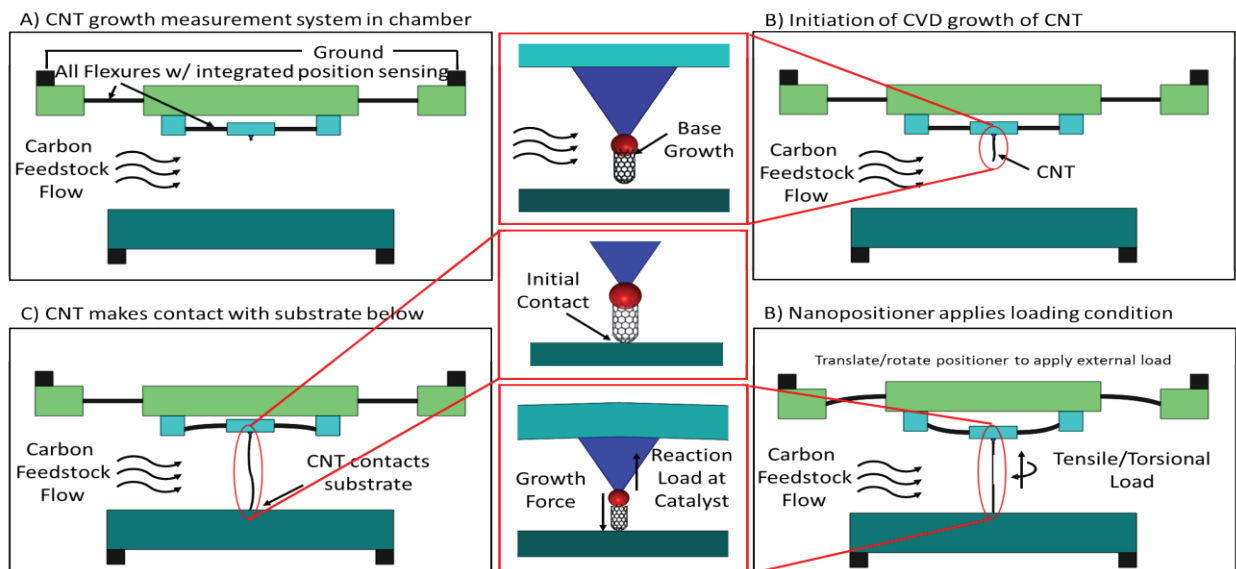


FIGURE 1. Schematic of growth force measurement process. A) Place force sensor into CVD chamber with catalyst on pyramidal tip and introduce feedstock. B) Activate micro heater to initiate CVD growth of CNT. C) CNT grows from force sensor to make contact with a separate substrate and measure voltage change to determine growth force. D) Use nano positioner to translate force sensor to introduce controllable loading conditions.

CNTs lifting tungsten weights [2]. This device produces a new method to analyze the CNT growth mechanism by allowing us to directly quantify the force generated by the CNT growing from a catalyst particle. Additionally, this system can provide an *in-situ* force control growth method, as shown in Fig. 1.d), to further test effects of growth force.

FORCE SENSOR

Design of the multi-axis MEMS force sensor is defined by a system noise model for piezoresistor MEMS sensors [3]–[5]. Figure 2 shows the three beam fixed-guided system coupled with a pair of piezoresistors at the base of each beam that compose the physical structure of the force sensor. These constraints allow displacement in the normal direction and rotation in pitch and roll of the central stage. The dimensions shown in Table 1 are determined by optimizing the signal-to-noise ratio in the system noise model. Designed force resolution is 71 pN which is less than half of the reported CNT growth force.

TABLE 1. Flexure and piezoresistors dimensions.

(μm)	Length	Width	Height
Flexure	3000	324.2	2
Piezoresistor	130.2	85.1	0.198

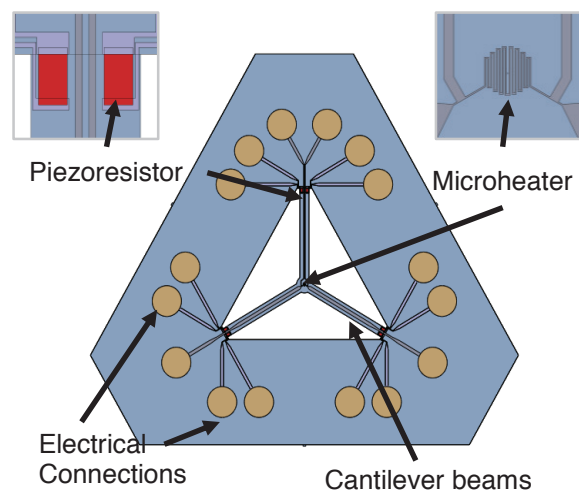


FIGURE 2. Multi-axis force sensor with integrated microheater model.

Integrated on the central stage of the force sensor is a microheater and a pyramidal tip. Tungsten is used as the microheater's material for its stability near the CVD growth temperatures

of approximately 900°C [6]. Additionally, two wires are added to create four point probe design for active monitoring of the heater's temperature. The pyramidal tip acts as a fixed position for catalyst placement and raises the growth initiation away from the surface.

FABRICATION PROCESS

The force sensor is fabricated using standard CMOS processes at the Microelectronics Research Center in Austin, Texas. A 100 mm double silicon-on-insulator (SOI) wafer is used as the substrate. The SOI wafer is composed of two 2 μm device layers and one 450 μm handle layer of silicon separated by two 0.5 μm silicon dioxide layers illustrated in first image in Figure 3. First, a 100 nm thermal oxide layer is deposited using a

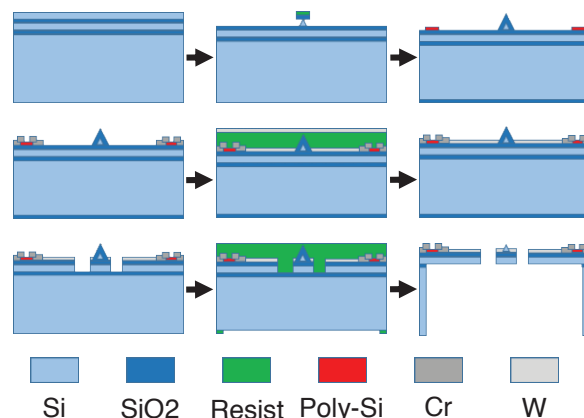


FIGURE 3. Process flow for the multi-axis force sensor fabrication.

low pressure CVD furnace, and 2 μm circular features are patterned using reactive ion etching (RIE). RIE replaced wet etching in order to preserve the oxide layer on the backside of the wafer. A 3 M KOH solution with saturated isopropyl alcohol (IPA) is used to define pyramidal structures into the top silicon device layer [7]. Next, 198 nm diffusion doped polysilicon layer is deposited with LPCVD with an n-type carrier concentration of 3×10^{19} atoms/cm³ and resistivity of 0.01 $\Omega \cdot \text{cm}$. Piezoresistors are defined with RIE. Chromium is deposited via electron beam deposition and etched with RIE to define the contact pads and wires for the piezoresistors. A tungsten microheater is patterned next using a liftoff process. RF sputtered Tungsten is deposited on patterned photoresist layer. Excess material is lifted off in an acetone ultrasonic bath resulting in the

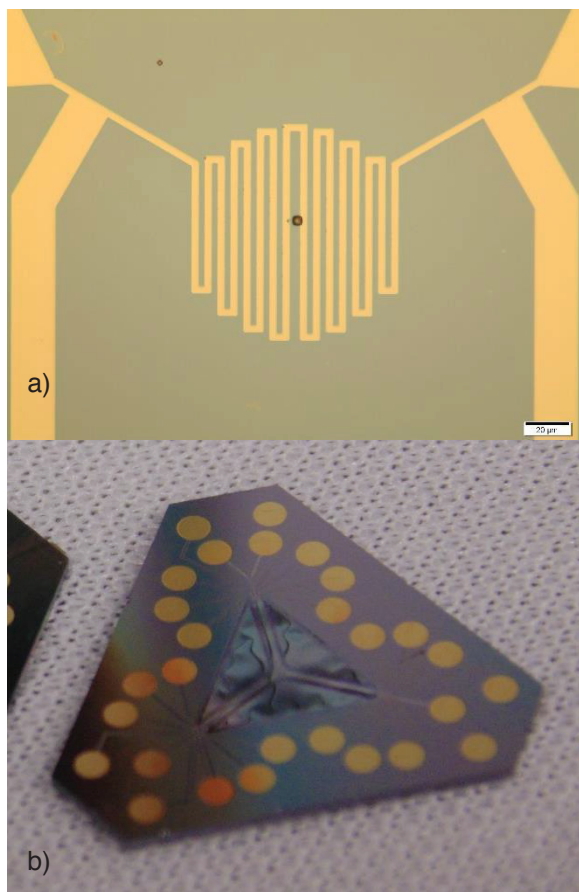


FIGURE 4. (a) Tungsten microheater around the pyramidal tip and (b) a single device after DSE backside etch. The beams are outlined by 2 μm silicon on 0.5 μm oxide diaphragm.

structure shown in Figure 4a). An Au/Cr stack is also deposited and patterned with liftoff to create wire bondable contact pads.

With all of the sensor's featured created, the cantilever beams can be defined and etched from the front side by RIE. In order to fully suspend the cantilever beams, the 450 μm handle layer is etched in a PlasmaTherm Versaline deep silicon etcher, Figure 4b). Two etching steps are used in this process. Etch A with RF Bias voltage of 150 V and pressure of 43 mTorr is designed to limit the amount of re-entrant over the depth of cut to preserve the fixed-end of the cantilever beam structures. The high pressure, 60 mTorr, and low RF Bias voltage, 10 V, in Etch B creates a high anisotropic etch while limiting 'grass' or black silicon formation [8]. A re-entrant of the sidewall is 3.7 degrees which created a 27.89 μm undercut and results in about a 1% error in the fabricated length of the cantilever beams. Once

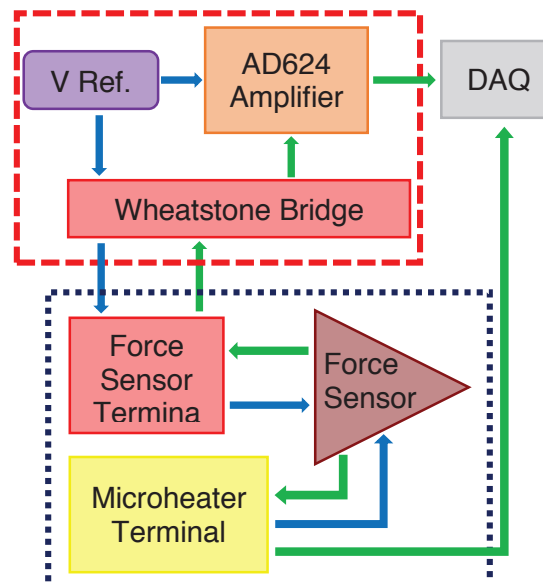


FIGURE 5. Block diagram of two circuit system. The precision amplifier circuit is outlined in red and the data collection circuit outlined in navy. The voltage/current supply flow is blue with the output in green.

the DRIE reached the oxide layer, an HF etch completes suspension process.

CIRCUIT BOARD DESIGN

Two circuits are implemented in the calibration process. A block diagram in Figure 5 highlights the main components and supply/output flow through the two circuits. The first circuit, shown in Figure 6 (a), is a precision amplifier circuit for data processing developed by DiBiasio et al. [9]. The main components are the 10 V precision reference voltage and the AD624 low-noise instrumentation amplifier. Both components are chosen to limit the amount of noise introduced into the measurement and to boost the signal-to-noise ratio. The voltage reference supplies 10 V to the Wheatstone bridge connected to each force sensor and acts as the reference voltage for the AD624. An adjustable gain arrangement is incorporated to maximize the resolution of the AD624.

The second circuit is the data collection circuit, shown in Fig. 6 (b). The creation of the data collection circuit is driven by the three Wheatstone bridges required per device. Two types of half bridge constructions are fabricated: one requiring external unstrained resistors and one with unstrained resistors fabricated on the sensor shown in Fig. 7. These factors led to a

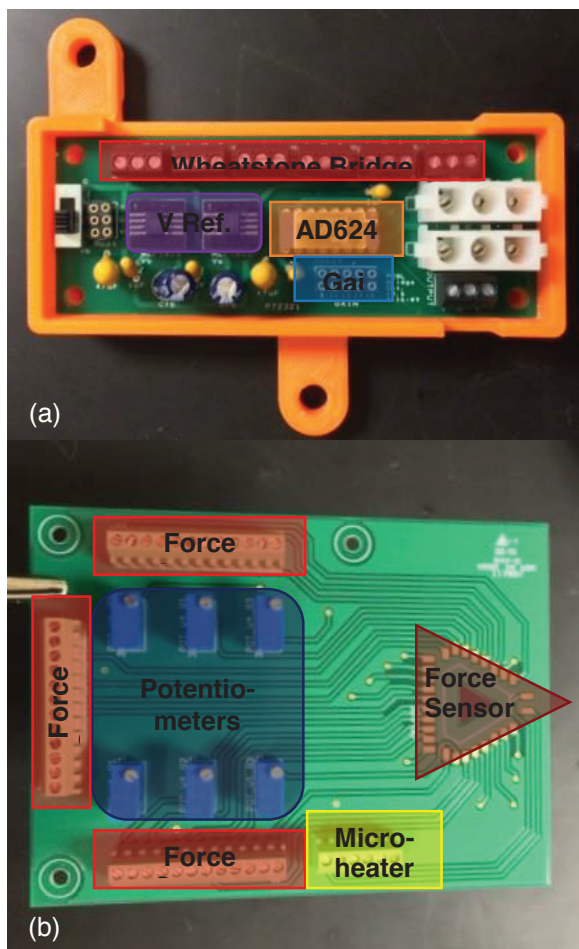


FIGURE 6. (a) Precision amplifier circuit in plastic housing. The highlighted features are: AD624 precision instrumentation amplifier – orange, precision voltage reference – purple, Wheatstone bridge – red, and adjustable gain – blue; (b) Data collection circuit. The highlighted features are: 12 pin terminal blocks for each force sensor – red, 6 pin terminal for microheater – yellow, cutout for device mount – maroon, and potentiometers – navy blue.

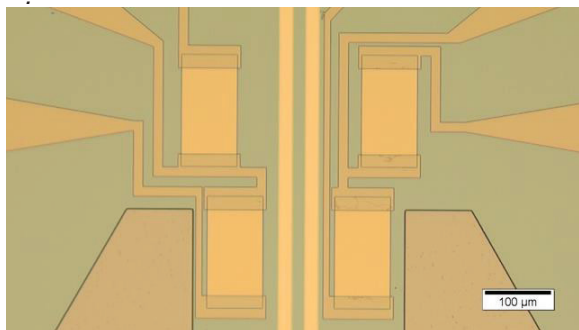


FIGURE 7: Force sensor with two unstrained polysilicon resistors fabricated on-chip.

design with three 12 pin terminal blocks where each block creates a half-bridge construction and connects to the data processing circuit. External resistors can be connected in their respective position on the data processing circuit. Potentiometers are incorporated to balance the bridge during null condition. Gold wire bonding connects the force sensor to the surface mount technology, SMT, pads on the circuit board. The force sensor is mounted over the cut region of the board using Crystalbond adhesive. Additionally, a 6 pin terminal block is used for the microheater and 4-point probe.

CALIBRATION

Calibration is necessary for both the force sensor and the microheater. The force sensor is calibrated using a Hysitron TI 950 TriIndenter operating in force control mode. A series of 0.5 μN indentations are executed around the central stage to map the stiffness and find the center of stiffness of the fixed-guided beam system. Next, indentations are run at the center of stiffness from 0.5 to 5.0 μN while recording the stiffness, displacement, and voltage response of the sensor. Stiffness and displacement are recorded to determine if the calibration is occurring in a linear region. The voltage can be plotted against force to create a calibration curve and factor for each sensor. Additionally, the gauge factor, GF, for the polysilicon piezoresistors can be determined by [10]:

$$GF = \frac{\Delta R}{R \cdot \epsilon} = \frac{\Delta V \cdot E_f \cdot b_f \cdot h_f^2}{6 \cdot i \cdot R \cdot F \cdot L_f} \quad \text{Eq. 1}$$

Where, ΔV is the change in voltage from unstrained to strained; E_f , L_f , b_f , and h_f are the beam's elastic modulus, length, width, and thickness respectively; R is the initial resistance of the piezoresistor; i is the current, and F is the applied load to induce the strain. The force sensor calibration will be repeated with devices across the wafer to determine the variations with respect to fabrication position and map potential calibration factors.

The focus of the microheater calibration is to define the temperature versus supply current curve and the resistance versus temperature curve. A range of currents is supplied to the microheater while a constant voltage is applied to the 4-point probe wires. A high resolution thermal camera, PSC-764-1M by Process Sensors, is

used to measure the temperature generated at each given current. The 4-point probe is an additional two wires connected to the beginning and end of the microheater to measure the voltage change. Ohm's law is used to convert the voltage change in a change in resistance. The resistance versus temperature calibration curve is generated. This curve can then be used to measure the temperature in-situ during the CNT growth experiments via change in resistance. Additionally, the temperature change across the piezoresistors is measured to be compared with previous simulations in order to validate our thermal models of the force sensor system [5].

CONCLUSION

The force sensor in this paper has a designed force resolution of approximately one half of the minimum anticipated growth force of a CNT. Calibration of the force sensor is used to determine a calibration factor between force and voltage, the polysilicon piezoresistive properties, and the calibration factor between the microheater's resistance and temperature. In the future, the calibrated force sensor will be used to study the growth forces of CNTs. This measurement is critical in advancing our understanding of the physics underlying the CNT growth mechanism.

REFERENCES

- [1] M. Kumar and Y. Ando, "Chemical Vapor Deposition of Carbon Nanotubes: A Review on Growth Mechanism and Mass Production," *J. Nanosci. Nanotechnol.*, vol. 10, no. 6, pp. 3739–3758, Jun. 2010.
- [2] A. J. Hart and A. H. Slocum, "Force output, control of film structure, and microscale shape transfer by carbon nanotube growth under mechanical pressure.," *Nano Lett.*, vol. 6, no. 6, pp. 1254–60, Jun. 2006.
- [3] R. M. Panas, M. A. Cullinan, and M. L. Culpepper, "Design of piezoresistive-based MEMS sensor systems for precision microsystems," *Precis. Eng.*, vol. 36, no. 1, pp. 44–54, 2012.
- [4] M. a Cullinan, R. M. Panas, and M. L. Culpepper, "A multi-axis MEMS sensor with integrated carbon nanotube-based piezoresistors for nanonewton level force metrology," *Nanotechnology*, vol. 23, no. 32, p. 325501, Aug. 2012.
- [5] I. S. Ladner and M. A. Cullinan, "Localized Growth and Force Detection of Carbon Nanotubes on Multi-axis MEMS Sensor," in *ASPE 2014 Ann. Meeting*, 2014, pp. 320–324.
- [6] M. S. Haque, K. B. K. Teo, N. L. Rupensinghe, S. Z. Ali, I. Haneef, S. Maeng, J. Park, F. Udrea, and W. I. Milne, "On-chip deposition of carbon nanotubes using CMOS microhotplates.," *Nanotechnology*, vol. 19, p. 025607, 2008.
- [7] I. Zubel, I. Barycka, K. Kotowska, and M. Kramkowska, "Silicon anisotropic etching in alkaline solutions IV," *Sensors Actuators A Phys.*, vol. 87, pp. 163–171, 2001.
- [8] A. J. Knobloch, M. Wasilik, C. Fernandez-pello, and A. P. Pisano, "Micro, Internal-Combustion Engine Fabrication with 900 um Deep Features via DRIE," *ASME*, pp. 1–9, 2003.
- [9] C. M. Dibiasio, "Concept Synthesis and Design Optimiztion of Meso-scale, Multi-Degree-of-Freedom Precision Flexure Motion Systems with Integrated Strain-based Sensors," Massachusetts Institute of Technology, 2010.
- [10] P. J. French, "Polysilicon : a versatile material for microsystems," *Sensors and Actuators*, vol. 99, no. A, pp. 3–12, 2002.

DESIGN OF A MEMS-BASED TUNABLE GRAPHENE RESONATOR SYSTEM WITH PRECISION STRAIN AND FORCE METROLOGY

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ABSTRACT

This paper presents the design of a MEMS testbed for controllably straining graphene resonators with precise load detection. By accurately controlling the initial tension on a single layer graphene resonator (GR), it is possible to tune the natural frequency of the resonator and to manufacture NEMS resonators that are more accurate, reliable, and repeatable than with current designs and manufacturing techniques. In this paper we present designs for 20 graphene resonator MEMS devices with different load and displacement resolutions (shown in Table 2). The load and displacement requirements of each MEMS device are set by the dimensions of graphene beam. The graphene resonator devices are will be fabricated at the wafer scale (4 inch diameter) to improve the cost effectiveness. Overall, there will be 40 units cell on a wafer with total number of 800 devices each capable of producing large strains of up to 15%-25% in the graphene resonators. The highest load and displacement resolutions of the test bed are 6.95 nN and 0.23 nm, respectively, from a 5 μm long graphene resonator with 1 μm width.

INTRODUCTION

Nanoelectromechanical (NEMS) resonators offer the potential to extend many of the limits in force and mass measurement due to their small size, high natural frequencies, and low energy dissipation (high quality factor or Q-factor). Graphene-based NEMS resonators are particularly promising due to their high elastic modulus ($\sim 1\text{TPa}$), high yield strain (up to 20%),

and atomic thickness [1]. However, the high volume, repeatable manufacturing of GRs is limited by large variations in natural frequency and Q-factor between devices. These large variations are caused by small changes in residual stress and initial tension of graphene film [2]. These variations can be eliminated by applying a uniform, constant tension to the GR. In order to controllably strain the GR, a microelectromechanical (MEMS) testbed with load detection will be used to apply tension to the GR. By controlling the tension on the graphene resonator, it should be possible to both the increase the repeatability between devices and to tune the natural frequency of the graphene resonators. The tunable graphene resonator testbed system could be useful in a number of areas including nanoscale force/mass detection in the biotherapy field, high-frequency signal processing, and in wireless communications applications [3].

CAD MODEL DESIGN

The MEMS-based tunable graphene resonator (GMEMS), as shown in Fig. 1(a), consists of a GR suspended between two movable electrodes attached to folded beams. The localized back gate is used as an electrostatic actuator to induce resonance in GR shown in Fig. 1(b), and two differential capacitors are used to measure the motion of both the electro-thermal actuator (ETA) and the load sensor (LS). The movable stage on left side is actuated using ETA which allows a uniform and constant tension to be applied to GR.

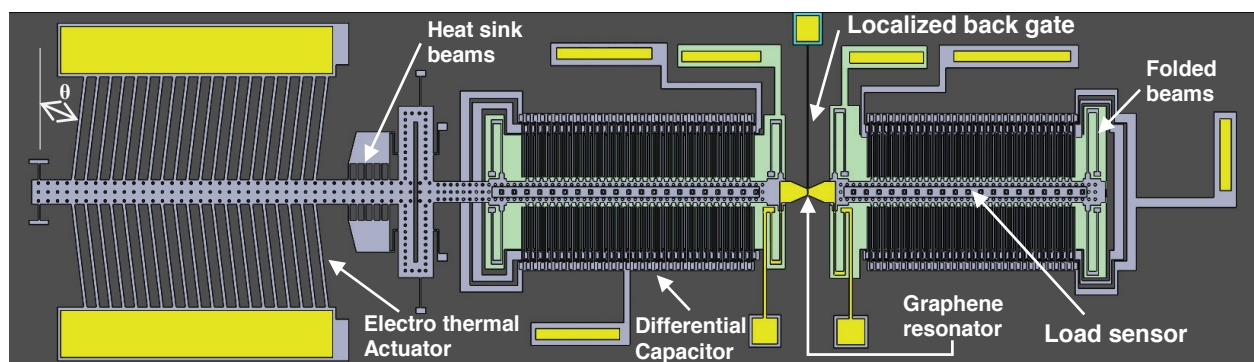


Figure 1(a). MEMS-based tunable graphene resonator system layout



Figure 1(b). Suspended graphene beam

A lumped model can be used to predict displacements of ETA, the specimen, and the load sensor as a first order approximation (Fig. 2) [4]. Compatibility of deformation and equilibrium leads to the following system of equations [4]:

$$\begin{aligned} X_{LS} + X_S &= X_A \\ K_{LS}X_{LS} &= K_S X_S \\ K_A X_A + K_S X_S &= F \end{aligned} \quad (1)$$

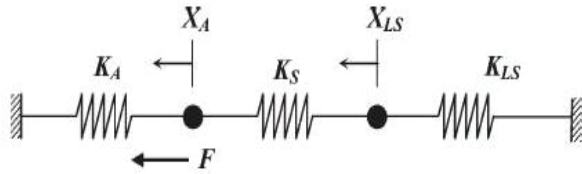


Figure 2. Schematic of lumped mechanical model. (X_S is the deformation of the specimen, X_{LS} is the relative displacement of the LS, X_A is the displacement of the ETA, K_S is the stiffness of the specimen, K_{LS} is the stiffness of the LS, K_A is the stiffness of the ETA, and F is the total force generated by the ETA.)

The GR stiffness, the load sensor (4 pairs of folded beams) stiffness, and the ETA stiffness with the addition of heat-sink beams and four pairs of folded beams are given by [4]:

$$K_S = \frac{EA}{l_S} \quad (2)$$

$$K_{fb} = \frac{2Eb_{ETA}^3 h_{ETA}}{l_{ETA}^3} \quad ; \quad K_{LS} = \frac{2Eb_{LS}^3 h_{LS}}{l_{LS}^3} \quad (3)$$

$$K_A = 2N \left(\sin^2 \theta \frac{Ebh}{l} + \cos^2 \theta \frac{Ehb^3}{l^3} \right) + 2n \frac{Ehb_{sb}^3}{l_{sb}^3} + K_{fb} \quad (4)$$

The force that is generated by the ETA is given by:

$$F = 2NEA\alpha\Delta T \sin\theta \quad (5)$$

Capacitance change for both LS and ETA is given by [4]:

$$\begin{aligned} C_0 &= N\epsilon_0 \left(\frac{A_1}{d_0} + \frac{A_2}{g} + 0.65 \frac{A_1}{h} \right); \quad C_3 = N\epsilon_0 \left(\frac{A_1}{d_3} + 0.65 \frac{A_1}{h} \right); \\ \Delta C &= 2NA_1\epsilon_0 \frac{1+2\frac{C_3}{C_0}}{d_0^2} \Delta d \end{aligned} \quad (6)$$

MULTIPHYSICS IN FEA

The performance of the MEMS stage in vacuum is simulated in ANSYS.14 by using a coupled field analysis involving electric, thermal, and mechanical fields. The input is the actuation voltage across the anchor sites and the output includes the temperature and displacement fields of MEMS actuator. In this multiphysics simulation, the polysilicon is selected as the material of the main mechanical structure with the properties as followed in table 1 [4]:

Table 1. Polysilicon Properties Used In The Multiphysics Simulations

Parameter	Units	Value
Young's modulus	GPa	170
Poisson's ratio	-	0.22
Thermal conductivity	W m ⁻¹ K ⁻¹	34
Resistivity	Ohm-m	3.4 × 10 ⁻⁵
Thermal expansion coefficient	K ⁻¹	2.6 × 10 ⁻⁶

From the previous simulation results, the displacement of the actuator increases as the number of inclined beams increases or the inclination angle, θ , decreases [5, 6]. Also, more heat sink beams lead to drastically reduced temperature at the specimen edge. However, the tradeoff of adding the heat sink beams is the decreasing efficiency in loading (or actuating) the specimen since the heat sink beams increase the overall stiffness of the structure.

In order to achieve an out-of-plane stiffness that is a few orders of magnitude greater than the in-plane stiffness of the GMEMS devices, a 6 micrometer thickness of the polysilicon layer is chosen. Also, 20 pairs of thermal beams at a 7° inclination angle and 10 pairs of 50 μm wide heat sink beams are found to offer the best performance. The thermal boundary conditions are zero temperature change at the anchors. The mechanical boundary conditions are fixed displacements at the anchor sites. Fig. 3 (a) and (b) shows the temperature distribution and the displacement respectively in the MEMS device for an operating actuation voltage of 3 V and a current of 4mA. In vacuum, the only heat dissipation path is through the anchors. Since the shuttle is the farthest from the anchors, the highest temperature occurs in the shuttle, as shown in Fig. 3 (a). As a result of the temperature nonuniformity in the structure, the displacement in the shuttle is also not uniform. As for out-of-plane motion of the MEMS, a maximum displacement of 35.78 nm from load sensor and of 75 nm from thermal actuator are shown in Fig. 3 (c). Overall, the simulation results agree well with calculations from the analytical models presented in the previous section.

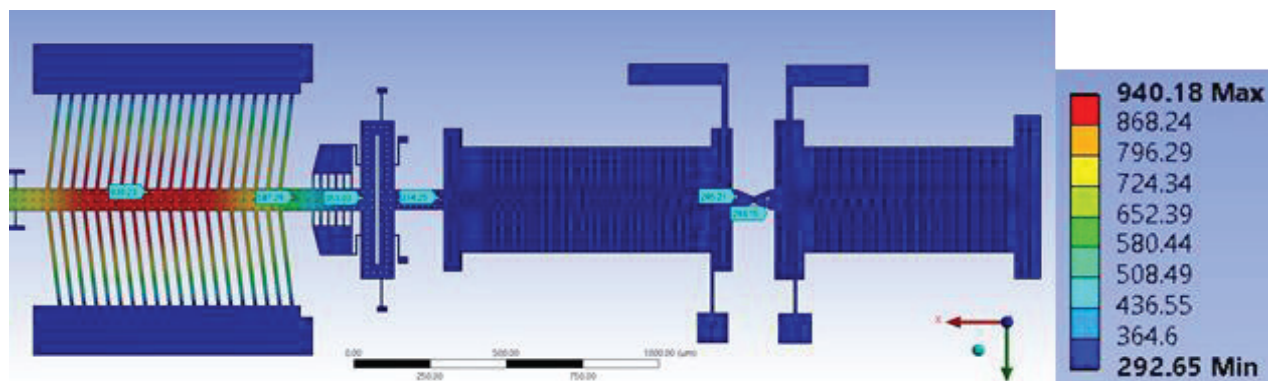


Figure 3a. Temperature (in K) fields in MEMS stage with a 2 μm long GR incorporated into the system

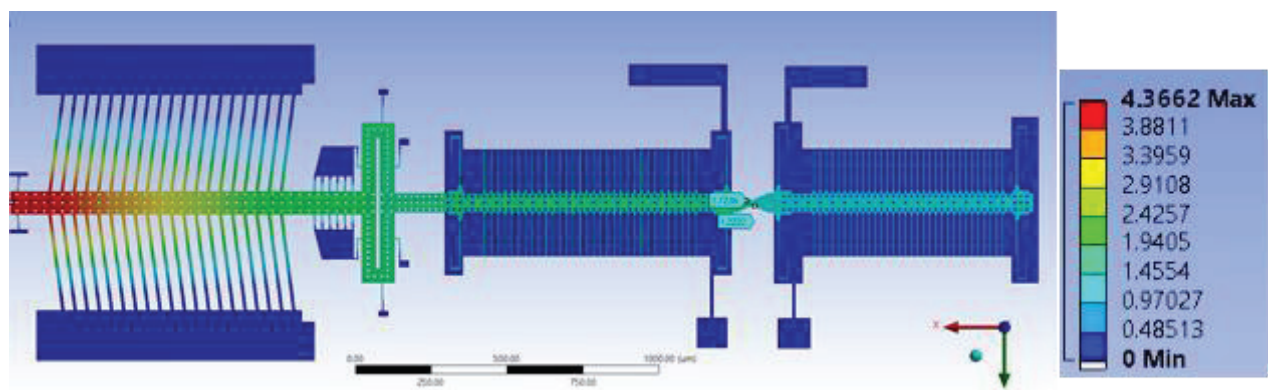


Figure 3b. Displacement (in μm) in X direction fields in MEMS stage with a 2 μm long GR incorporated in the system

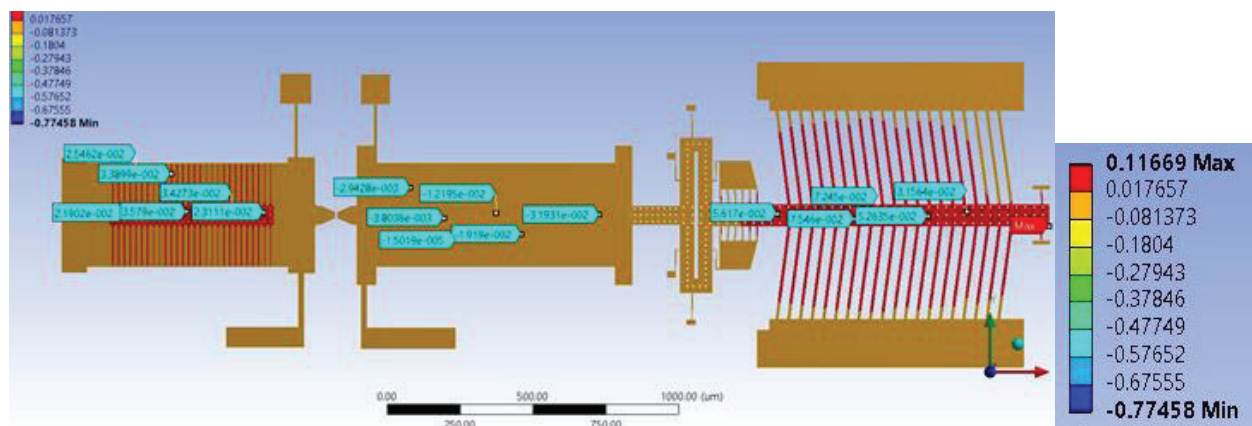


Figure 3c. Displacement in Z direction (out of paper) (in μm) for the MEMS stage with a 2 μm long of GR incorporated in the system

RESULTS

Table 2. Operating Parameters and Resolutions of Four MEMS Stages Incorporating Different Size of GR

GR dimensions		Operating parameters					capacitive sensitivity; Resolution in Load and displacement				
GR length	GR Width	Strain (%)	ΔT (K)	K_S (N/m)	K_{LS} (N/m)	K_A (N/m)	Load (nN)	$\Delta d(LS)$ (nm)	ΔC (LS) (fF/nm)	ΔC (ETA) (fF/nm)	$\Delta d(ETA)$ (nm)
1 μ m	1 μ m	25	274	335	164.96	2.763E4	37.82	0.23	0.22	0.11	0.45
	2 μ m	25	300	670	280.48	2.763E4	64.30	0.23	0.22	0.11	0.45
	5 μ m	25	320	1675	604.44	2.763E4	138.57	0.23	0.22	0.11	0.45
	10 μ m	25	310	3350	1341.3	2.763E4	307.5	0.23	0.22	0.11	0.45
2 μ m	1 μ m	25	647	167.5	45.21	2.763E4	10.36	0.23	0.22	0.11	0.45
	2 μ m	25	647	335	130.56	2.763E4	29.93	0.23	0.22	0.11	0.45
	5 μ m	25	647	837.5	309.47	2.763E4	70.94	0.23	0.22	0.11	0.45
	10 μ m	25	647	1675	604.44	2.763E4	138.57	0.23	0.22	0.11	0.45
5 μ m (high strain)	1 μ m	25	647	67	71.9	2.763E4	16.48	0.23	0.22	0.11	0.45
	2 μ m	25	647	134	144.53	2.763E4	33.13	0.23	0.22	0.11	0.45
	5 μ m	25	647	335	364.78	2.763E4	83.62	0.23	0.22	0.11	0.45
	10 μ m	25	647	670	743.44	2.763E4	170.43	0.23	0.22	0.11	0.45
5 μ m (high resolution)	1 μ m	15	647	67	30.33	2.763E4	6.95	0.23	0.22	0.11	0.45
	2 μ m	15.8	647	134	64.24	2.763E4	14.72	0.23	0.22	0.11	0.45
	5 μ m	16	647	335	164.96	2.763E4	37.81	0.23	0.22	0.11	0.45
	10 μ m	15.7	647	670	322.19	2.763E4	73.86	0.23	0.22	0.11	0.45
10 μ m	1 μ m	16.2	647	33.5	66.31	2.763E4	15.20	0.23	0.22	0.11	0.45
	2 μ m	16.1	647	67	130.5	2.763E4	29.93	0.23	0.22	0.11	0.45
	5 μ m	16	647	167.5	322.19	2.763E4	73.86	0.23	0.22	0.11	0.45
	10 μ m	16.1	647	335	669.15	2.763E4	153.41	0.23	0.22	0.11	0.45

FABRICATION PROCESS

The standard poly-MUMPs fabrication process is followed to build the device structures. There are 20 devices arranged in each unit, and 40 units arranged on a wafer. The process begins with 100mm n-type (100) silicon wafer of 1-2 Ω -cm resistivity. The surface of wafers are first heavily doped in diffusion furnace by using POCl_3 as dopant, which helps to prevent charge feedthrough to the substrate from electrostatic devices on the surface. Then, a low stress nitride layer of 600nm thickness is deposited on silicon wafer by LPCVD. A blanket 0.5 μ m layer of undoped polysilicon as ground layer is deposited using LPCVD followed by deposition of phosphosilicate glass (PSG) which is annealed at 1050°C for 1 hour. The polysilicon layer is then

lithographically patterned and dry etched using reactive ion etching (RIE). The anneal serves to both dope the polysilicon and reduce its residual stress. Next, an oxide layer is deposited by LPCVD and is lithographically patterned as a sacrificial layer between the first poly layer and second poly layer. This layer is then etched by RIE.

A 6 μ m polysilicon layer is selected as the second poly layer to act as the major mechanical structure in the GMEMS devices. This layer is deposited by LPCVD using the same processing conditions as the first polysilicon layer. After the main mechanical structure is lithographically patterned and etched by RIE, 350nm of silicon nitride is deposited by LPCVD as electrical

insulation layer between polysilicon and metal layer.

Next, a 200 nm copper layer is deposited using electron beam deposition and patterned as sacrificial layer to suspend graphene beam. A graphene film is then transferred onto the copper layer. The graphene is then lithographically patterned and dry etched by using an oxygen plasma. In the future, graphene transfer-free fabrication technique will be utilized to improve the manufacturing effectiveness [7].

When all mechanical structures have been fabricated, the remaining steps are to deposit a gold metal layer using the lift-off process, release the copper layer to suspend the graphene structures, and, finally, remove the sacrificial layer of oxide. Since the fabrication work is still ongoing, the first layer of poly and oxide have been fabricated and they are shown in the Fig. 4(a), (b) and (c). From the Fig. 4(a), the number marker that is located on left upper corner stand for each unique device as respect to different dimension of graphene beam, load and displacement resolution. That marker will be very helpful in collecting the reliability between devices as regard to their position on the wafer, and for making improvements in next generation of devices.

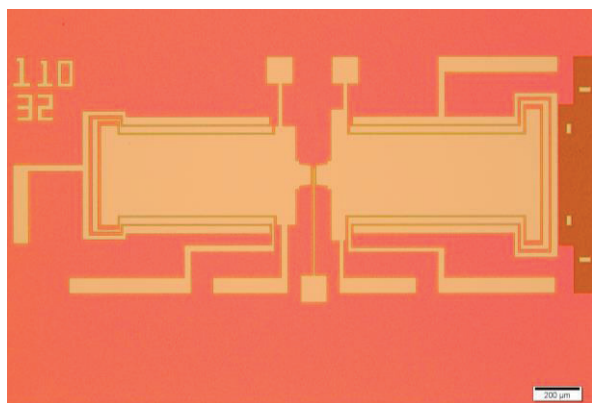


Figure 4(a) First layer of poly and oxide in 200 μm scale

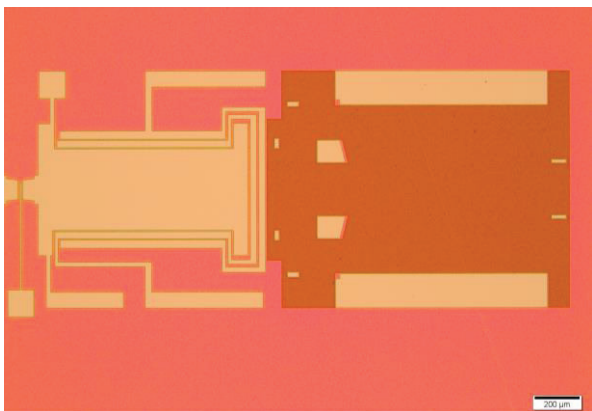


Figure 4(b) First layer of poly and oxide in 200 μm scale

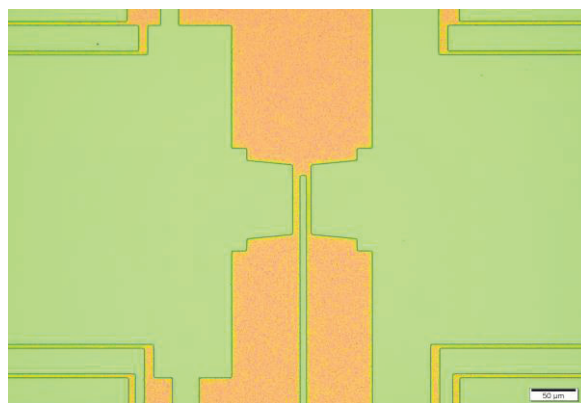


Figure 4(c) Zoomed localized back gate and two end of electrode in 50 μm scale

CONCLUSION AND FUTURE WORK

The GMEMS stage design developed in this paper is the first step towards achieving more precise, accurate, and repeatable graphene resonators. The rest of planned fabrication process is to be completed in near future. Importantly, the GMEMS is an ideal testbed for achieving several goals in the future: 1) Improving the manufacturability of graphene resonators by incorporating the wafer scale production of graphene into a MEMS microfabrication process, 2) Measuring the reliability, repeatability and accuracy of this new graphene manufacturing process, 3) Increasing the resolution of graphene-based force and mass detectors by application of uniform tension to the graphene film, 4) Determining the effect of the quality of the graphene/electrode interface on the Q-factor of the graphene resonator, 5) Assisting in improving our fundamental understanding of the mechanics and the electromechanical properties of graphene.

REFERENCES

- [1] C. Lee, X. Wei, J. W. Kysar, and J. Hone, "Measurement of the elastic properties and intrinsic strength of monolayer graphene.," *Science*, vol. 321, no. 5887, pp. 385–8, Jul. 2008.
- [2] Yuehang Xu, Changyao Chen, Vikram V. Deshpande, Frank A. DiRenno, Alexander Gondarenko, David B. Heinz, Shuaimin Liu, Philip Kim, and James Hone, "Radio frequency electrical transduction of graphene mechanical resonators", *Appl. Phys. Lett.* 97, 243111(2010)
- [3] D. Garcia-Sánchez, A. San Paulo, M. J. Esplandiú, F. Perez-Murano, L. Forró, A. Aguasca, and A. Bachtold, "Mechanical detection

of carbon nanotube resonator vibrations”, *Phys. Rev. Lett.*, vol. 99, no. 8, p. 85501, 2007.

[4] Horacio D. Espinosa, Yong Zhu, and Nicolaie Moldovan, “Design and Operation of a MEMS-Based Material Testing System for Nanomechanical Characterization”, *JOURNAL OF MICROELECTROMECHANICAL SYSTEMS*, VOL. 16, NO. 5, OCTOBER 2007

[5] Y. Zhu, A. Corigliano, and H. D. Espinosa, “A thermal actuator for nanoscale *in situ* microscopy testing: Design and characterization”, *J. Micromech. Microeng.*, vol. 16, no. 2, pp. 242–253, Feb. 2006.

[6] Qingquan Qin and Yong Zhu, “Temperature control in thermal microactuators with applications to in-situ nanomechanical testing”, *APPLIED PHYSICS LETTERS* 102, 013101 (2013)

[7] M Cullinan, JJ Gorman, “Transfer-free, wafer-scale fabrication of graphene-based nanoelectromechanical resonators”, *Microsystems for Measurement and Instrumentation (MAMNA)*, 2013, 3-6

ACTIVE OPTICS IN THE MANUFACTURE OF 8.4 M MIRRORS FOR TELESCOPES

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USE OF ACTIVE OPTICS IN MANUFACTURE OF TELESCOPE MIRRORS

Manufacture of large optics requires analysis of the bending of the mirror due to changing forces and temperature gradients. To form sharp images, a telescope mirror such as a segment of the Giant Magellan Telescope (GMT), Figure 1, must be accurate to around 50 nm rms on large scales. This accuracy is rarely achieved during the manufacturing process, because the large-scale shape is very sensitive to mirror support forces and temperature gradients in the glass. The final accuracy is not achieved until the mirror is in the telescope, and then only with active control of the 165 support actuators and feedback from a wavefront sensor. This process is known as active optics [1]. In addition to bending the primary mirror, active optics includes adjusting the alignment of the mirror(s) to minimize certain low-order aberrations.

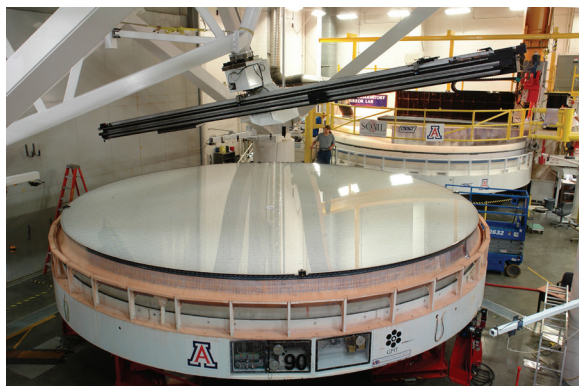


FIGURE 1. GMT Segment 1 at the Richard Caris Mirror Lab, University of Arizona.

During manufacture, it is simpler and safer to polish a mirror on a passive support system rather than the active system that will be in place at the telescope. While the passive support is designed to match the nominal support forces at the telescope, there are inevitably force differences of tenths of a percent, which can bend the mirror by hundreds of nm in astigmatism and other low-order modes.

Furthermore, drifts in support forces and temperature gradients cause significant shape changes even in a controlled environment, and these cannot be corrected with the passive support system.

We deal with this manufacturing issue by simulating the active optics correction: we measure the mirror surface in the lab, and compute optimized actuator forces and the residual surface error after optimizing. The relation between forces and shape changes comes from a finite-element model of the mirror [2]. The residual error is the best estimate of the mirror's shape with active optics working at the telescope, and it must meet a tight specification. The critical requirements for manufacture are (1) reduce the large-scale surface errors to the level that can be corrected with active optics, and (2) reduce the small-scale errors to the level that meets the wavefront specification.

APPLICATION TO GMT SEGMENTS

This simulation of active optics is standard practice in the manufacture of telescope mirrors. But large telescopes often have unique requirements related to active optics. That is the case for the GMT and the Large Synoptic Survey Telescope (LSST). The GMT is a 25 m telescope being built at Las Campanas Observatory in Chile [3]. Its primary mirror comprises seven 8.4 m segments: a central symmetric segment and six identical off-axis segments.

Active optics for the GMT involves two novel aspects. First, all seven segments must have essentially the same radius of curvature after the active optics corrections. This means the focus aberration (a symmetric paraboloidal surface error) must be corrected well, unlike a traditional symmetric mirror. Second, alignment plays a strong role in the active optics. Small displacements of the off-axis segment along the parent asphere cause low-order aberrations. Two types of motion—radial displacement and

clocking about the center of the segment—give two additional degrees of freedom for control of the wavefront. These alignment degrees of freedom can be used to reduce the amount of bending required and the associated actuator forces and glass stresses [4].

Figure 2 illustrates the simulated active optics process applied to a lab measurement of GMT Segment 1. The first step is to optimize the alignment of the segment in the telescope, and the second step is to bend out the remaining error with the actuators. Alignment has a limited effect on the magnitude of surface error because only two degrees of freedom are available to control 5 aberrations. It does, however, have a significant effect on the final result, both the residual surface error and the magnitude of force required to bend the mirror. This is because, when the alignment is optimized, the affected aberrations are weighted in proportion to the difficulty of bending them out. In Figure 2, the alignment corrects the focus well and largely ignores the astigmatism, because it is much easier to bend out astigmatism. The second step, bending the mirror, is very effective at removing the remaining error.

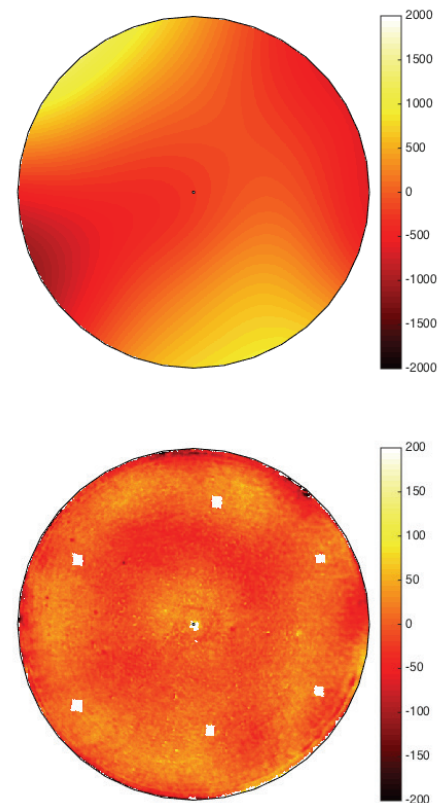
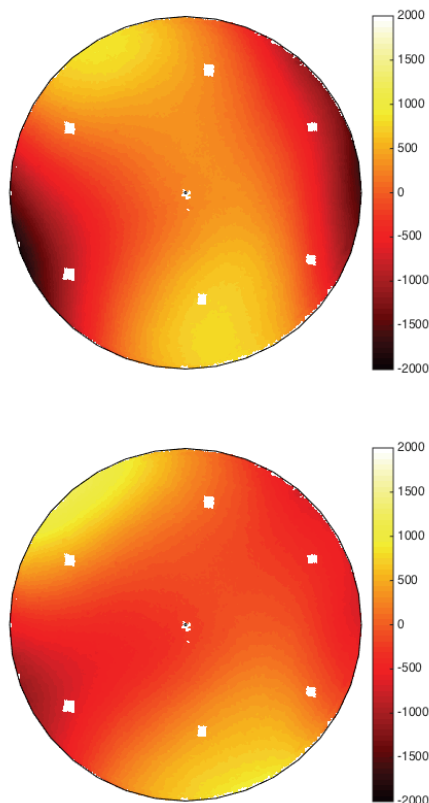


FIGURE 2. Illustration of the simulated active optics correction of GMT Segment 1. From top to bottom: measured surface error (583 nm rms); error after optimized alignment (437 nm rms); best fit of mirror bending (15 N rms force over 165 actuators); residual error after bending (23 nm rms). Color bars show surface error in nm.



In theory one could do even better with the 165 actuators. In practice the process is more predictable and robust if we use a limited number of low-order bending modes, 20 modes in Figure 2. The bending modes are a set of shape changes that can be produced by the actuators [1,2]. Each actuator produces a particular shape change, its influence function. The bending modes are orthogonal linear combinations of the influence functions, ordered from the most flexible combination to the stiffest. Using the first 20-30 bending modes gives the most improvement for a given magnitude of correction forces and makes the process work better in practice.

Figure 3 shows a few of the most flexible bending modes. Each mode is the result of a set of forces with a magnitude of 1 N rms over the

165 actuators. The magnitude of surface deflection decreases as we move toward stiffer modes. Generally, it is easy to produce shape changes fit by the lower bending modes, and difficult to create the higher modes.

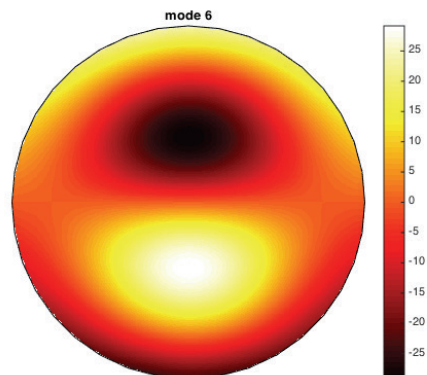
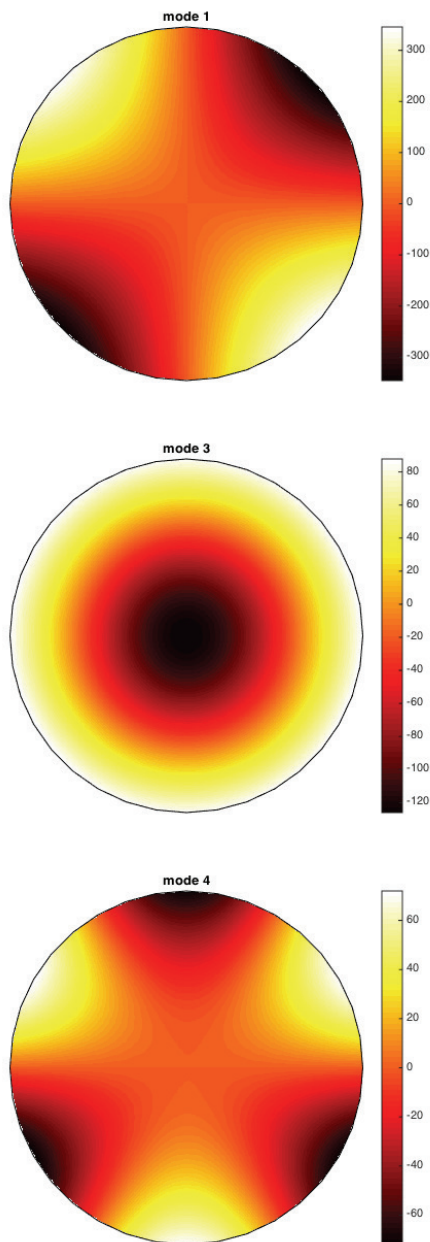


FIGURE 3. Four of the most flexible bending modes for the GMT off-axis mirror. Non-axisymmetric modes come in orthogonal pairs of about the same flexibility; the two members of each pair are rotated relative to each other. Color bars show surface deflection in nm for 1 N rms force.

Lab measurements of a telescope mirror do not use the mirror's natural property of reflecting collimated light to a point focus, because we cannot generate a plane wavefront over a large diameter. The wavefront measurements made at the telescope, which do use this property, are more accurate on large scales than the lab measurements. This is especially true for the GMT off-axis mirrors. The low-order aberrations are very sensitive to alignment of the various components of any test. We therefore employ two independent tests for low-order aberrations and require agreement between the tests.

At the time we completed Segment 1, we had excellent agreement for all aberrations except spherical aberration, the symmetric shape error with a quartic dependence on r . One test gave an acceptable 40 nm rms surface error in spherical, and the other gave a problematic value of 100 nm rms. The discrepancy was larger than the predicted measurement uncertainty, and the larger value appeared to be at the edge of what could be corrected by bending once accurate measurements are made at the telescope. Using the active optics recipe illustrated in Figure 2, in which the spherical aberration is corrected by bending, the best we could achieve was 29 nm rms surface error with 39 N rms force, and the bending caused stress in the glass that significantly reduced the margin of safety.

We investigated both tests and, after several months, found a subtle error that reduced the larger value from 100 nm to 50 nm, making the discrepancy insignificant. During the period of uncertainty, there was great interest in knowing whether the mirror would perform well if the larger result for spherical aberration was correct. Simply bending out 100 nm rms spherical required high forces and produced high stress in the glass. It turns out, however, that the mirror's first axisymmetric bending mode shown in Figure 3 (mode 3) is nearly a combination of focus and spherical aberration in the ratio 7:1. It is relatively easy to bend the mirror in this way and produces little stress. Focus can be controlled both by alignment (a radial displacement) and by bending. We therefore modified the active optics recipe to shift the mirror by 0.9 mm, introducing 700 nm rms focus, and bend out the combination of focus and spherical aberration.

Figure 4 shows the simulated correction, modified to correct spherical aberration. The first step of the correction, optimized alignment, includes the radial shift and focus change. The resulting shape error is larger but can be corrected more easily and more accurately by bending. The residual surface error is almost identical to that in Figure 2 and the correction forces are only slightly larger. In the end, with agreement between the two tests, we do not need to use the modified recipe, but it illustrates an interesting way to make use of the alignment degrees of freedom in the active optics correction.

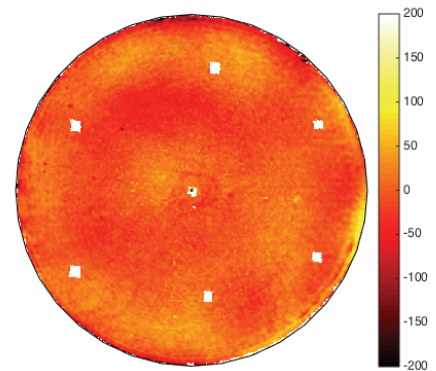
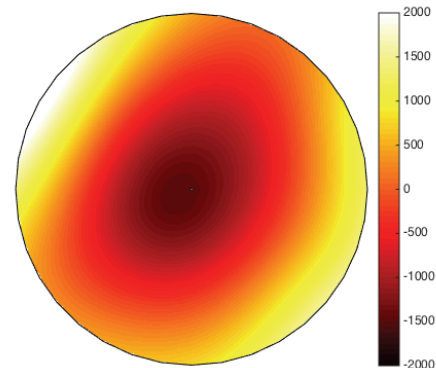
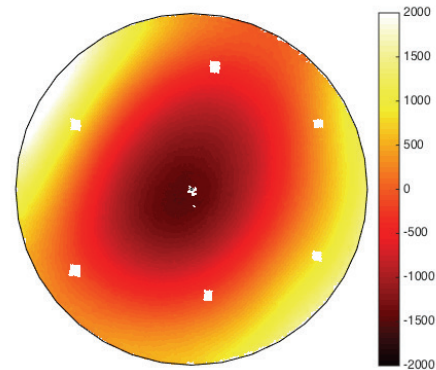
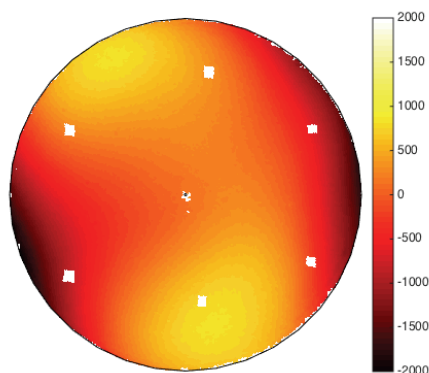


FIGURE 4. Simulated active optics correction of GMT Segment 1, with the recipe modified to correct 100 nm rms spherical aberration. From top to bottom: measured surface error (589 nm rms); error after optimized alignment (811 nm rms); best fit of mirror bending (17 N rms force); residual error after bending (23 nm rms). Color bars show surface error in nm.

APPLICATION TO THE LSST MIRRORS

The LSST is a three-mirror telescope with an 8.4 m primary mirror (M1), a 3.4 m secondary mirror and a 5.1 m tertiary mirror (M3) [5]. During the design process it became clear that putting the

primary and tertiary mirrors on the same piece of glass offered several advantages. It eliminated 5 alignment degrees of freedom that would otherwise have to be controlled in the telescope, and it made M1 stiffer as part of a complete disk rather than a separate annulus. The final design made M1 the outer annulus $2.53 \text{ m} < r < 4.20 \text{ m}$, and M3 an inner annulus $0.53 \text{ m} < r < 2.53 \text{ m}$. Figure 5 shows the combined LSST mirror.

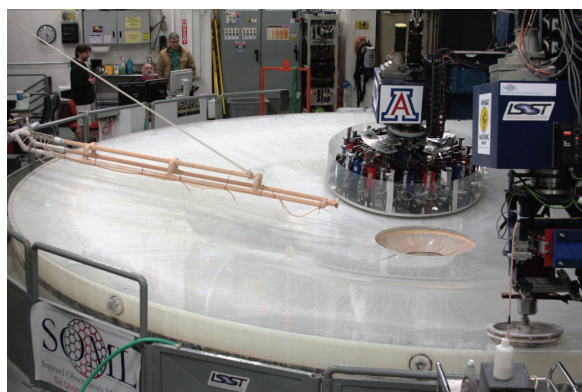


FIGURE 5. Combined LSST primary and tertiary mirror on the polishing machine at the Mirror Lab. The smaller tool in the foreground is polishing M1 while the larger tool polishes M3.

The combined mirror shares a single set of 156 support actuators as well as the glass. The two mirrors cannot be bent independently; whatever forces optimize the shape of M1 also change the shape of M3. When we simulated the active optics correction, it was necessary to show that both mirrors could be brought to accurate shapes (within about 30 nm rms) with the same set of actuator forces. One implication of this additional requirement is that the measurements of both mirrors had to capture the glass in nearly the same state, i. e. with nearly identical support forces and temperature gradients. Simultaneous measurements were possible, but M3's test optics had to be located between M1 and its test optics and therefore compromised the M1 test. To obtain the most accurate measurements of both mirrors, we used a nearly simultaneous "1-3-1" test, measuring M1 then M3 then M1 at equal intervals. We averaged the two M1 measurements to eliminate first-order drifts in shape due to changing forces and temperatures. We were able to show that the 1-3-1 test was equivalent to a simultaneous test within the specifications.

Showing that both mirrors can be brought to nearly perfect shapes brings in additional

constraints compared with simulating active optics with a single mirror. The allowed shape of one mirror is tightly constrained by the shape of the other. M1, being the outer mirror, is more sensitive to support forces, so the active optics has more leverage on M1. While much of the polishing was done to both mirrors in parallel, we finished M1 first and made use of the active optics correction to achieve an accurate large-scale shape. This meant we could not use the simulated active optics freely to correct M3. The strategy for figuring M3 was therefore: (1) apply the bending forces that optimize the shape of M1; (2) compute the resulting shape error on M3; (3) polish that error out of the M3 surface.

Figure 6 illustrates the active optics correction of the combined mirror and shows that both mirrors are brought to $< 30 \text{ nm rms}$ surface error with a single set of correction forces. The data for the two mirrors are obtained separately (with the 1-3-1 test) but are shown in a single image to better illustrate the active optics correction. For the symmetric LSST mirrors, unlike the GMT off-axis segment, alignment is decoupled from bending in the active optics correction. The active optics correction only involves bending the combined mirror.

We chose to give full weighting to M1 in the active optics correction. We computed the shape change for M3 but did not let it influence the force optimization. The recipe can be tuned to give equal weight to both surfaces or any other relative weighting, and this might be done at the telescope. There is little difference in results between full weighting of M1 and equal weighting of the two mirrors, because M1 is so much more sensitive to the support forces.

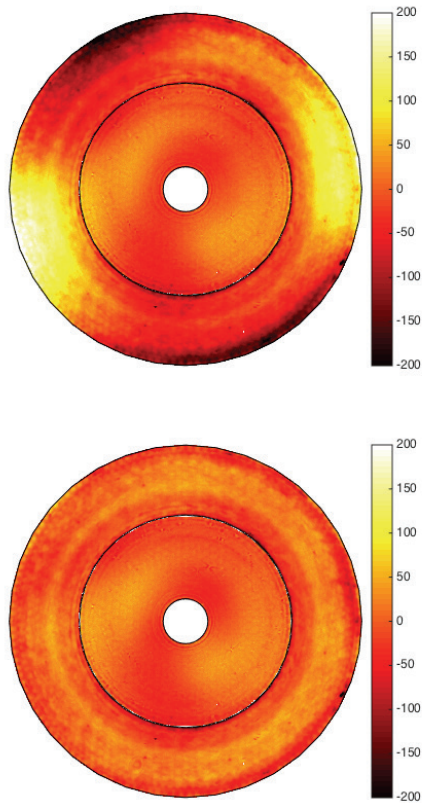


FIGURE 6. Simulated active optics correction of combined LSST primary and tertiary mirror. Top: measured error (M1: 77 nm rms; M3: 28 nm rms). Bottom: residual error after bending (M1: 29 nm rms; M3: 23 nm rms). Color bars show surface error in nm.

SUMMARY

We modified the standard active optics recipes to work with GMT off-axis segments and with the LSST combined primary-tertiary mirror. As part of the manufacturing process, we simulated the active optics correction, and used the results to show that these mirrors will meet all requirements and deliver excellent images at the telescope. We expect that the modified active optics recipes will be used at the telescopes to maintain excellent wavefront accuracy.

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REFERENCES

- [1] Wilson RN. Reflecting Telescopes II: Manufacturing, Testing, Alignment, Modern Techniques. Berlin: Springer; 1999.
- [2] Martin HM, Cuerden B, Dettmann LR, Hill JM. Active optics and force optimization for the first 8.4 m LBT mirror. In: Oschmann JM, Tarenghi M, editors. Ground-based Telescopes. Proceedings of SPIE 5489. 2004; p. 826.
- [3] Bernstein RA, McCarthy PJ, Raybould K, Bigelow BC, Bouchez AH, Filguera JM, et al. Overview and status of the Giant Magellan Telescope project. In: Stepp LM, Gilmozzi R, Hall H, editors. Ground-based and Airborne Telescopes V. Proceedings of SPIE 9145. 2014; p. 91451C-1.
- [4] Burge JH, Kot LB, Martin HM, Zehnder R, Zhao C. Design and analysis for interferometric measurements of the GMT primary mirror segments. In: Atad-Etchedgui E, Antebi J, Lemke D, editors. Optomechanical Technologies for Astronomy. Proceedings of SPIE 6273. 2006; p. 6273OM-1.
- [5] Gressler W, DeVries J, Hileman E, Neill DR, Sebag J, Wiecha O, et al. LSST telescope and site status. In: Stepp LM, Gilmozzi R, Hall H, editors. Ground-based and Airborne Telescopes V. Proceedings of SPIE 9145. 2014; p. 91451A-1.

FABRICATION OF AN IR TELESCOPE FOR A BALLOON-BORNE SPACE MISSION

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INTRODUCTION

The NASA Balloon Experimental Twin Telescope for Infrared Interferometer (BETTII), shown in FIGURE 1, consists of two identical telescopes mounted onto a space frame and carried to an altitude of 40 km by a balloon [1]. Each telescope has four optical elements – a primary mirror, a turning flat, a secondary mirror, and a tertiary mirror.

The balloon-based mission brings with it many unique challenges for the design, fabrication and assembly of the optical system. During the ascent into the high atmosphere, the telescope will experience temperature changes of 120 °C. To minimize the effect of these thermal cycles, the entire telescope is Aluminum 6061-T651 so that the frame and the optics will all expand and contract the same amount.

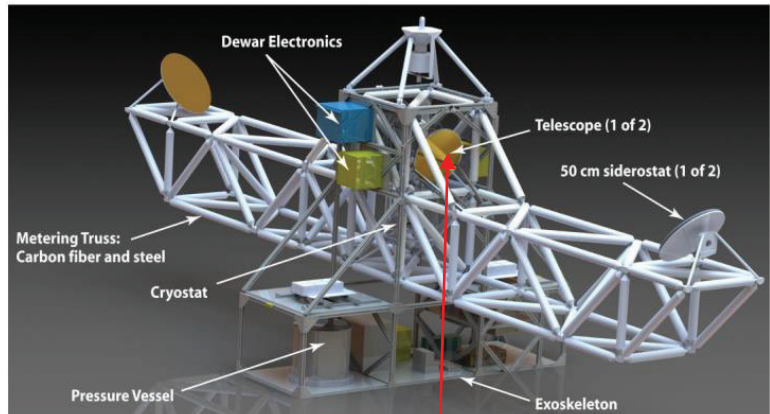


FIGURE 1: BETTII Infrared Binocular Space Telescope [1]

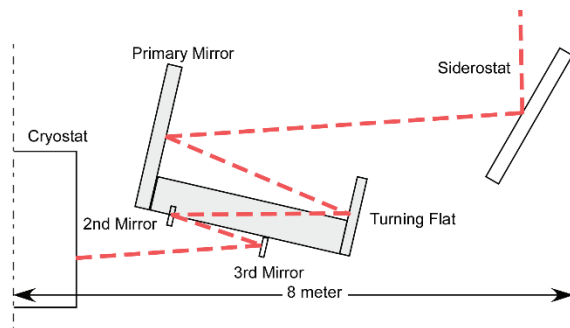


FIGURE 2. Layout of the telescope optical system

Figure 2 shows the layout of one of the telescopes. The siderostat can be used to follow the star group of interest and send that image to the off-axis parabolic telescope primary, the flat turning mirror and two additional focusing and turning optics. This optical train sends a reduced aperture, collimated beam into the cryostat where it is combined with an identical telescope on the opposite side of the 16 m wide space frame. The Cryostat with the optical sensors is hung from the two telescopes which in turn are mounted through flexures to carbon fiber/steel truss. The telescope is tilted 13.3° with respect to the horizontal to allow full optical access to the primary mirror from the siderostat.

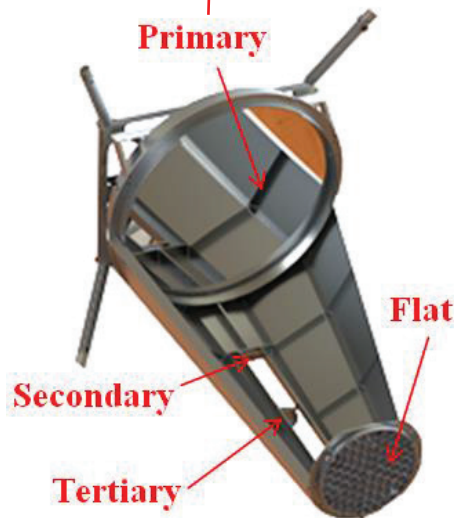


FIGURE 3. Structure of the telescope showing the primary (back) and flat turning mirror (front)

Details of each telescope assembly are shown in Figure 3. The large off-axis parabolic primary is a segment of a 2.7 m parabolic mirror that is 652 mm off-axis with an aperture of 522 mm. The flat mirror at the other end turns the beam back to the secondary mirror.

TELESCOPE STRUCTURE

The primary and the smaller flat mirror are connected with a circular “trough” including rings on each end that mate with the mirrors (Figure 4). The trough also supports the two smaller mirrors at the bottom (Figure 4, parts 4 and 5) that send the beam into the Cryostat [2].

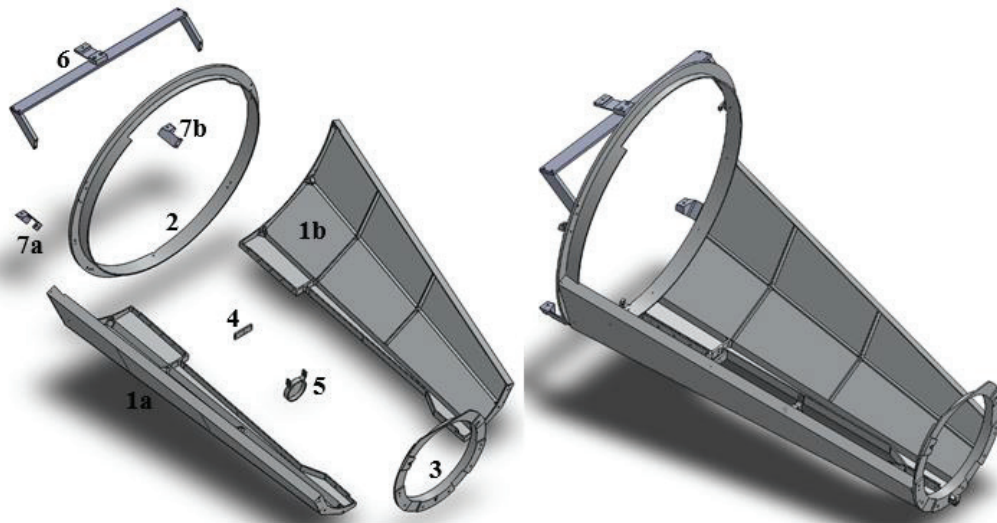


FIGURE 4: Mounting structure with thin-walled trough and coupling rings

RINGS AND TROUGH DESIGN

The primary mirror has a 560 mm diameter with a mass of 5.7 kg and the flat turning mirror has a 250 mm diameter and a mass of 0.64 kg. The mirrors are both mounted to rings. Both of the rings have an L-shaped cross-section machined from a solid plate of aluminum and includes 3 radial v-shaped grooves that support toroidal features machined on the face of each mirror.

To reduce the manufacturing complexity, the trough was split into two parts along the center line of the telescope and each was machined from a single block of aluminum. What started as a 180 kg block became one-half of the 3.4 kg trough. The two pieces are bolted together along the center line and to the rings on each end that support the mirrors.

The surface area, volume, and mass of each component within the structure was carefully considered. It is important to keep the surface area to volume ratio – and thus thermal time constant – similar among the components. All of the mirrors,

mounting brackets and trough had a surface area to volume ratio between 0.31 and 0.99 mm²/mm³. After assembly, thermally conductive paste and tape will be applied to speed conductive heat transfer between the thin-walled trough and the thicker components.

KINEMATIC KELVIN COUPLING AND CLAMP

The mirrors are attached to the rings using a kinematic Kelvin coupling that enables repeatable positioning and decouples deformations in the mounting structure from the optical surface. The Kelvin coupling consists of a toroidal feature that is on the front of each mirror and a triangular groove in the mounting rings, as shown in Figure 5. Having these mounting features on the front of the mirror allows them to be finished with the same tool that is used to rough the optical surface, enabling the mirrors to be precisely located. The mirrors are held on by clamps that provide a constant, pre-determined load through the center of the toroid features.

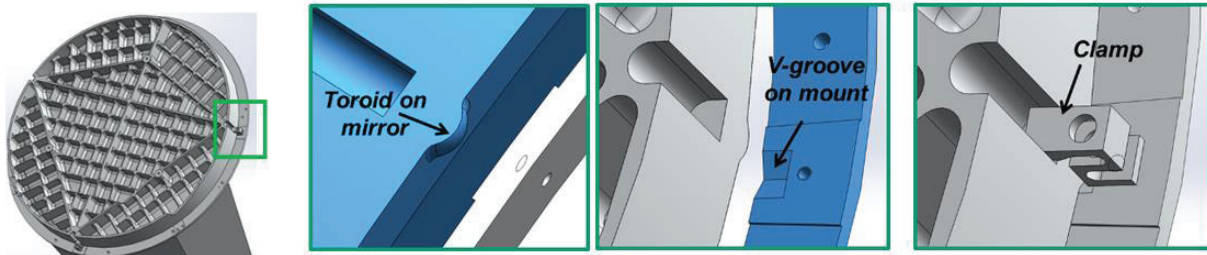


FIGURE 5: Kinematic Kelvin coupling

The dimensions and stiffness of the custom clamps and mirror structure were measured to get an

estimate of the clamping load applied. Figure 6 shows the clamp. The total height of the clamp, h ,

and the thickness of the contacting pad, t , were measured and the undeflected clamp height, h_0 , was calculated as the difference. All 12 clamps had undeflected heights between 13.24 and 13.29 mm. The distance between the contact surface on the back of the mirror and the mounting ring to which the clamps were bolted, h_1 , was measured using a CMM. These heights ranged between 13.70 and 13.73 mm, causing a range of clamp deflections between 0.41 and 0.49 mm (design was 0.45 mm).

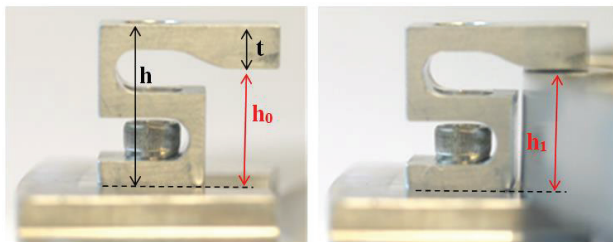


FIGURE 6: Custom fixed-load clamps for kinematic Kelvin coupling, undeflected (left) and holding the flat mirror (right).

The clamp's stiffness was measured by bolting the clamp to an optical table and measuring deflection using an electronic indicator while pushing on the clamp with a force gage. The stiffness was constant at 0.176 N/ μ m for multiple applied loads up to 83 N. Therefore the clamp loads are expected to range between 72.2 and 86.2 N, compared to the designed load of 85 N. The clamps were easy to install and produced repeatable loading.

ASSEMBLY OF MOUNTING STRUCTURE

All of the designed parts were machined and assembled. The difference in length of the two sides of the 811 mm long trough was determined to be 0.09 mm by measuring the step height on the flat end. All bolts in the structure are 10-32 316 stainless steel socket cap screws with 316 stainless

nuts. They were tightened to 45 N-m of torque. The assembled mounting structure is shown in Figure 7.



FIGURE 7: BETTII telescope mounting structure

MIRROR DESIGN AND FABRICATION

FLAT MIRROR DESIGN

The flat mirror is a light-weighted structure with an OD of 254 mm, thickness of 12.7 mm and a weight of 0.64 kg. The flat mirror has the three toroid-shaped features on its face to mate with the features on the ring at the front of the mounting structure.

The back of the flat mirror is light-weighted using a single depth pattern shown in

Figure 8. This design combats the gravity sag from its 13° tilt in the telescope as well as print-through during the diamond turning operation. The features on the back side of the flat mirror can be cut with a square end mill that eliminates the weight-adding fillets at the bottom of the pockets.

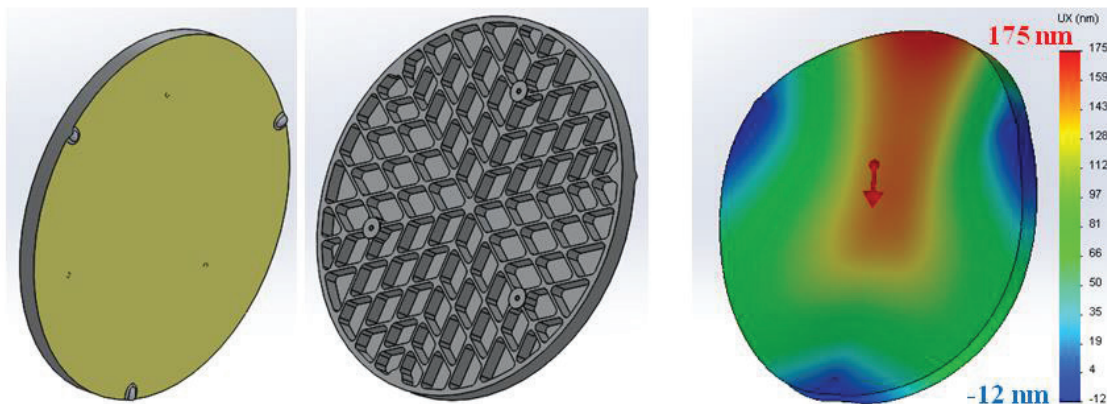


FIGURE 8: Front (left) and back (middle) of flat turning mirror. Deflection due to gravity sag (right).

The outside diameter of the flat mirror is 254 mm and it is 12.7 mm deep. The face is 2 mm thick, the outer rim is 4 mm wide, and the light-weighting ribs are 2 mm wide, making the mirror weigh 0.64 kg. The raised bosses on the back are lapped flat and used for mounting during diamond turning.

MACHINING FLAT MIRROR

The mirror was bolted to the spindle of a NanoForm 600 DTM through three, flat bosses on the back of the mirror. These bosses were threaded for a 1/4-20 stud to connect them to the diamond turned face of the chuck. The stiffness of the mirror was measured with a cap gage and load cell and ranged between 0.7 – 1.5 N/ μ m depending on proximity to the mounting boss. The flat mirror was machined at 200 rpm with a crossfeed of 25 μ m/rev, and the final figure error was 0.175 μ m rms with a surface finish of 30 nm rms. There was no indication of print-through of the light-weighting features on the face of the 2.5 mm thick face of the mirrors from the interferometer measurements.

Shape/Surface finish

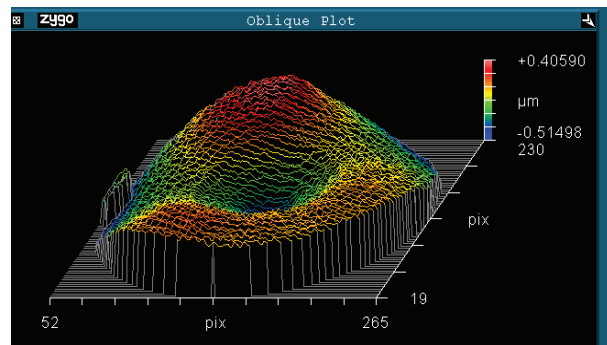


FIGURE 9 Form error: PV 921 nm, RMS 175 nm

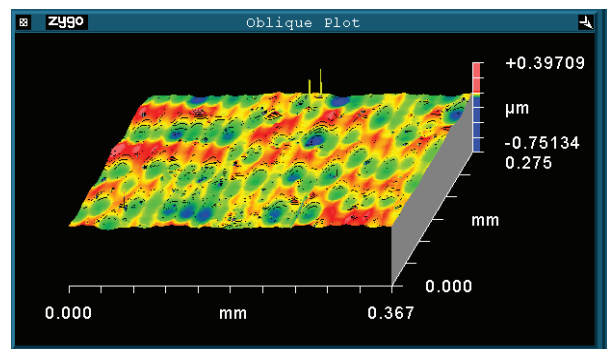


FIGURE 10 Surface finish: RMS 30 nm

Performance of Kinematic Coupling

The primary role of the Kelvin coupling is to mount the thin, flexible optics to the mounting structure without imparting a moment to the optical surface. To test this, the flat mirrors were measured on a Zygo GPI white-light interferometer while sitting on

top of 3 pegs on the back-side of the mirror, opposite the toroid mounting features. Then the stiff, L-shaped flat ring was attached to the flat using the Kelvin coupling with the custom clamps (Figure 6), and the flat was measured again. A comparison between the measurements is shown in Figure 11 and demonstrates that no significant change occurred in the surface shape, despite the three ~85 N clamp loads. Both interferometer measurements show three high spots at the location of the mounting supports and low spots in between where the thin mirror has sagged due to gravity. The RMS surface of the flat bolted to the mounting ring (bottom) is 163 nm.

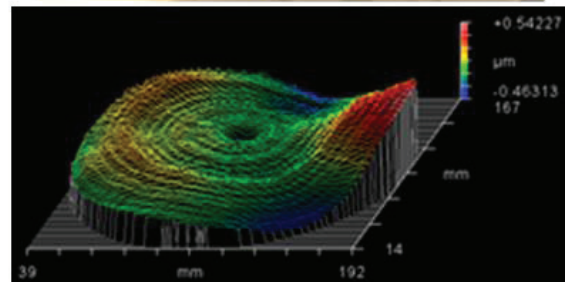
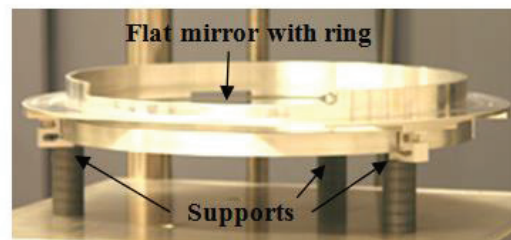
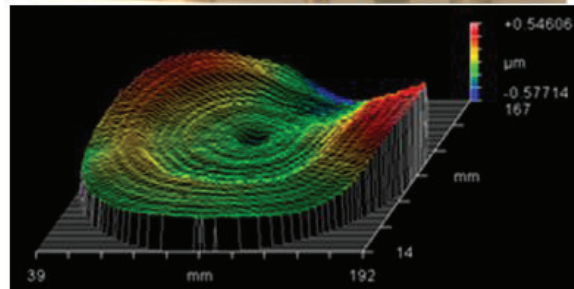
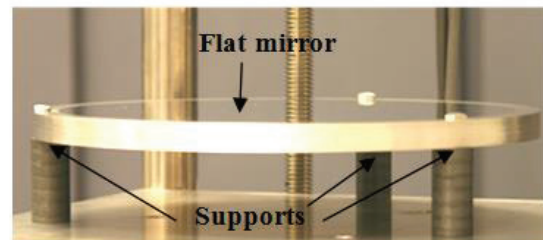


FIGURE 11: Surface form error of flat mirror unmounted (top) and clamped in kinematic mount (bottom)

PRIMARY MIRROR

Primary Mirror Design

The primary mirror is a segment of a 2.1 m parabola that is 652 mm off-axis with an aperture of 522 mm. It, like the flat, was designed with light-weighting ribs and mounting bosses on the back and toroidal features on the front that form half of the non-influencing kinematic mount.

The light-weighting features consist of thicker ribs with a wide spacing to combat gravity sag and thin ribs spaced 30 mm apart to reduce the amplitude of print-through during diamond turning. This mirror is 50 mm thick with a mass of 5.7 kg [2].

Machining Process

Because of the size and the lack of rotational symmetry, the mirror must be machined on a DTM using a fast tool servo (FTS). In this case, a Nanoform 600 DTM was coupled with the FLORA II fast tool servo built at the PEC [3]. The spindle speed is 60 rpm to avoid significant centrifugal distortion. The final finishing pass requires 3 hours, during which time the oil flow and chip vacuum are continuously managed to prevent loose chips from degrading surface finish. The tool path for machining the off-axis parabolic section on axis is created by translating the off-axis aperture to the center of rotation, tilting it and subtracting the best fit rotationally symmetric surface. For an off-axis parabola the best fit on-axis surface is another parabola. The resulting sag is 13 mm and this is the path that the DTM axes follows from the OD to center. The 1 mm correction for the non-rotationally symmetric surface, shown in Figure 3 is simultaneously added by the FTS.

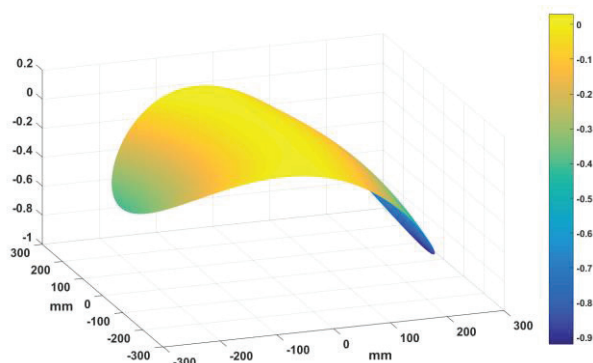


FIGURE 12. NRS component of the primary mirror

FLORA II Programming and Operation

FLORA II is a voice-coil-driven, air-bearing FTS with a range of 8 mm that can move the required 1 mm excursion for the primary mirror at 60 rpm with a following error of ± 22 nm. However that performance is predicated on a complex control

system that includes position feedback and velocity and acceleration feed-forward. The controller update rate is 20 kHz, so every 50 μ sec new position, velocity and acceleration set points are calculated. The control system uses an Analog Devices SHARC 21061 32-bit floating point DSP.

Following a technique developed at the PEC, an analytic solution to the decomposition of the off-axis paraboloid into an on-axis rotationally symmetric shape and a non-rotationally symmetric residual was found [4] and the solution implemented using VisualDSP++. The velocity and acceleration feed forward terms were calculated off-line as functions of r and Θ with balanced finite differencing and then used to find the coefficients of best-fit sinusoidal surfaces. On each update cycle the controller uses these coefficients to estimate the required feed forward set points.

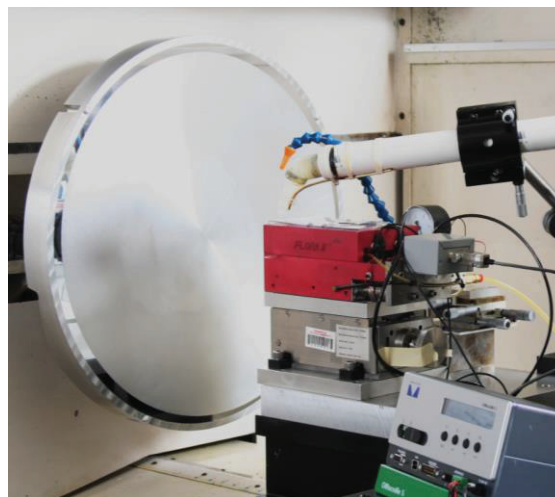


FIGURE 13. Roughed primary mirror with diamond-turned flat perimeter mounted on the DTM with FLORA II

FLORA Tool Location and Replacement

When cutting the large primary mirror, each roughing pass will involve 2 km of cutting distance and the finishing pass 9 km. Since 8-10 roughing passes may be needed to clean up the part, it is desirable to replace the tool between roughing and finishing processes. Previous measurements with this round-nose tool indicated that the tool edge is expected to recede by 700 nm in Z and edge radius increase to 750 over the first 12 km of cutting, as shown in Figure 14. The SEM EBID technique for measuring tool wear was developed by Arcona and Dow at the PEC, and used extensively by Lane [5] and Hessler [6]. In Figure 14, the flank face is on the right, rake face is on the left, and worn edge is oriented vertically. The red dots are placed on the deposited EBID contamination line and compared

to the profile of the sharp tool, with the area shaded in green representing the worn area of $2.16 \mu\text{m}^2$.

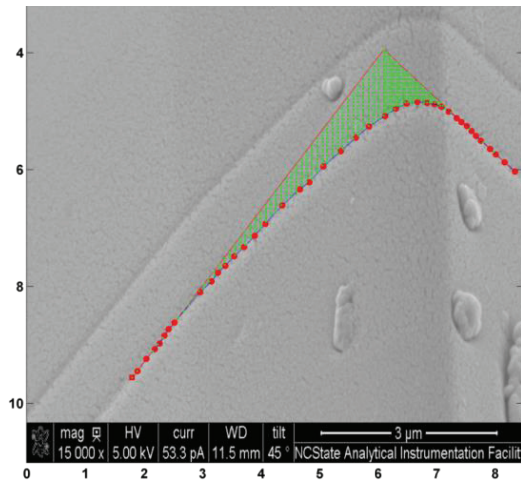


Figure 14. SEM image showing the edge of a diamond tool worn by 12 km of cutting AL 6061.

The large, non-axially symmetric part cannot be removed and precisely replaced while the tool is changed, so the new tool must be located at the spindle centerline without using the common method of cutting a spherical test plug. To accomplish this task, a new method was developed to locate the nose of the tool with a confocal chromatic aberration (CA) probe. The CA probe used, Precitec CHRocidile S, RB200 031, has a $5 \mu\text{m}$ diameter spot and 10 nm vertical resolution. The probe is mounted to the x-axis as shown in Figure 15 and is used to scan the top of the diamond.

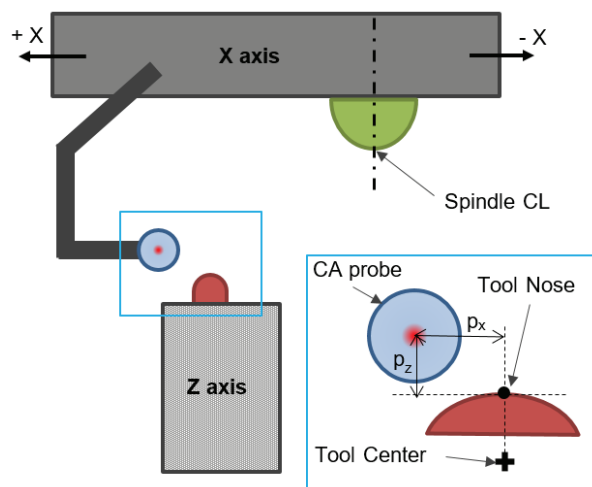


Figure 15. Schematic showing the spindle and probe mounted to the X axis and the tool to the Z.

The user jogs the axes to a position just off the nose of the tool, indicated by the G55 CS in Figure 16. Then the scan makes 9 traces across the tool in Z

and the data is filtered to eliminate points that are not on the top of the diamond. A circle is fit to the edge points of the diamond and the center points and radius of this circle locates the nose of the tool. In the scan in Figure 16, the fit circle has a radius of 1.010105 mm, center of $\langle -0.01404, -1.09305 \rangle$, and the average edge height of 0.03929 mm.

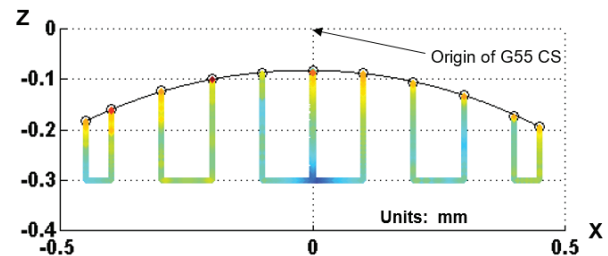


Figure 16. CA probe measurements of tool 1.

When the new tool is mounted, the scan in Figure 16 is repeated using the same fixed probe and deviations in the tool location are calculated. Deviations are corrected by first raising FLORA on the micro-height adjuster, then re-scanning to find the X and Z positions again. Finally, the X and Z offsets on the machine are adjusted so that the new tool will be aligned with the spindle centerline. The process was successfully verified by cutting test plugs before and after the tool replacement. The repeatability of the tool location measurement was validated by taking 8 measurements on the same tool. Given the same scan starting position, the standard deviation of the X, Z and height measurements were 58 nm, 71 nm, and 15 nm respectively. When varying the starting position of the scan within a $35 \mu\text{m}$ range, the deviations increased to 538 nm, 200 nm, and 37 nm.

References

- [1] S. Rinehart. Balloon Experimental Twin Telescope for Infrared Interferometry. Proc. SPIE, 7734, 2010.
- [2] S. Furst, T. Dow, K. Garrard. Design and Fabrication of an Optical System for Balloon-Borne Space Telescope. Proc. ASPE, 196-199, 2014.
- [3] E. Zdanowicz. Design of a Fast Long Range Actuator – FLORA II. Ph.D Thesis North Carolina State Univ. 2009.
- [4] K. Garrard, T. Bruegge, J. Hoffman, T. Dow and A. Sohn. Design tools for freeform optics. Proc. SPIE, 5874, 95-105, 2005.
- [5] BM Lane, Material Effects and Tool Wear in Vibration Assisted Machining. PhD Thesis North Carolina State University 2012.
- [6] GS Hesler. Investigating the Mechanisms of Diamond Tool Wear Cutting Ferrous Materials Using a Quantitative Study of Machining Parameters. MS Thesis, North Carolina State University.

PRODUCTIVE FREEFORM MACHINING USING A LONG STROKE FAST TOOL SYSTEM

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INTRODUCTION

Freeform optics are increasingly being utilized in optical systems for the aerospace, automotive, sensor, LED lighting, and consumer electronics industries to improve overall system performance [1], [2], [3]. The diverse use of freeform optics requires a wide range of geometries. Combiner or other internal mirrors for HUD systems are often large (about 100 x 200 mm²) but have smooth surfaces with low dynamic requirements. Conversely, optics for LED illumination and automotive headlights require smaller but more complex freeforms with higher dynamic demands [4]. With this growing need for freeform optics, manufacturers are increasingly using ultra-precision diamond machining to fabricate prototype optics and molds for high volume replication processes.

The non-rotationally symmetric geometry of freeform surfaces requires dynamic axes for the tool movement to generate the shape through diamond machining. Commercially available and established fast tool systems offer such dynamics but are limited in stroke length. Hence, manufacturers often have to use the more time consuming slow tool process, as the resulting stroke of the overall tool path spiral exceeds the limit of today's fast tool systems. Slow tool machining not only requires longer machining times (up to days) and higher costs but also limits the accuracy of the freeform surface due to the influence of environmental effects. Drifts caused by temperature variations of the surrounding, can influence the work piece accuracy.

The new long stroke fast tool system on the UPC 400 ultra-precision center by Schneider overcomes these limitations by offering more than 20 mm of stroke. This ensures greater flexibility during machining of a wide range of freeforms and shorter process times while increasing productivity and accuracy. This paper will discuss the advantages of a dynamic, long stroke fast tool system in comparison with slow tool machining.

Both processes are integrated onto the UPC 400 (see FIGURE 1). The UPC 400 combines diamond machining, milling, and metrology onto a single platform to enable the manufacturing of freeform optics up to 400 mm in diameter.

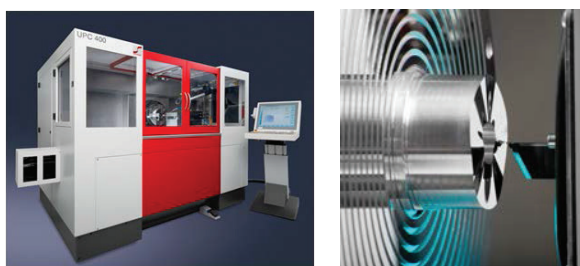


FIGURE 1. UPC 400 with close-up of the fast tool system cutting a sinusoidal mirror.

DISCUSSION

Fast tool systems are highly dynamic tool positioning devices that are used to machine non-rotationally symmetric surfaces like freeform geometries. During the rotation of the work piece the tool position is varied along the Z-axis to follow the given freeform surface. Fast tools can achieve high dynamics since essentially, only the tool with a low mass is moved during the machining process. In most cases fast tools are used in addition to the standard X and Z linear axes.

The first fast tools were based on a piezo drive in combination with flexural bearings and offered only a few micrometers of stroke (see FIGURE 2, left). The initial aim of these short stroke systems was to increase the machine tool accuracy by compensating errors of linear axes or spindles [5], [6]. But this type of fast tool was also used to machine non-rotationally symmetric surfaces like faceted laser mirrors [7].

Later, the ophthalmic industry drove the demand for systems with a stroke of several millimeters for the machining of spectacle lenses. Therefore systems based on rotary and linear axis systems were developed [8], [9] (see FIGURE 2).

Today, long stroke fast tools that are established in the field of ultra-precision machining primarily utilize voice coil drives and linear air-bearing slides with a stroke below 7 mm.

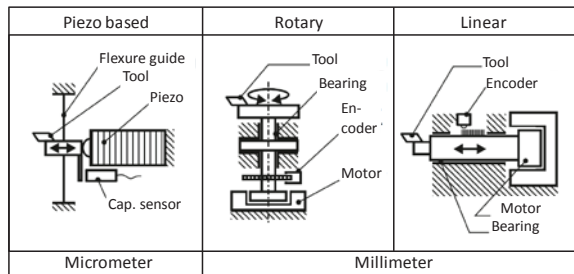


FIGURE 2. Principle design of different fast tools systems according to [10].

In contrast, slow tool machining uses the Z-axis of an ultra-precision lathe. Hence no additional system is integrated onto the machine. During slow tool machining basically the whole stroke of the machine slide is available which can be hundreds of millimeters. However, since the entire slide has to be moved, the dynamic range is very limited and can be further reduced if several devices (i.e. tools, B-axis, milling spindle, etc.) are mounted on top of the slide. This can lead to enormous machining times during freeform manufacturing limiting the throughput and work piece accuracy.

For high productive freeform machining the UPC 400 includes a fast tool with a stroke length that exceeds current systems by 400%. In addition, the fast tool can achieve acceleration rates greater than 35 g.

Therefore this system enables a very flexible and productive freeform machining process. Due to its long stroke it can be used to machine freeforms with large excursions from the best-fit sphere. One example is the off-axis machining of HUD molds as shown in FIGURE 3. The concave mold insert for a combiner screen has a sagittal depth of 9 mm and outer dimensions of about 100 x 200 mm².

Furthermore it has to be considered, that the overall stroke during freeform machining is depending on two things. First, the peak-to-valley value or at least the non-rotationally symmetric portion of the given freeform surface defines the minimum required stroke. In addition, during off-axis machining or if rectangular base materials are used the tool path outside the work piece has

to be considered for the overall stroke as well. Outside the workpiece the machine software automatically extrapolates the tool path to close the spiral, while strictly keeping the tool path, for several derivations, steady. This avoids the generation of vibrations when the tool re-enters the cutting zone on the work piece. Nevertheless this procedure often leads to large amplitudes in the tool path, especially if the slope at the work piece edge is very steep. These amplitudes very often cannot be achieved using currently commercially available fast tool systems. The new long stroke fast tool supports very homogenous tool paths with large amplitudes for various manufacturing tasks.

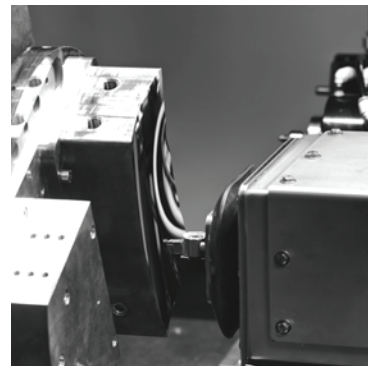


FIGURE 3. Off-axis machined HUD mold insert using a long stroke fast tool.

In addition, the acceleration capability of the fast tool used in the UPC 400 allows for the machining of complex, highly dynamic structures, as well. FIGURE 4 shows the machining of a faceted mirror having a sagittal depth of 100 μ m.

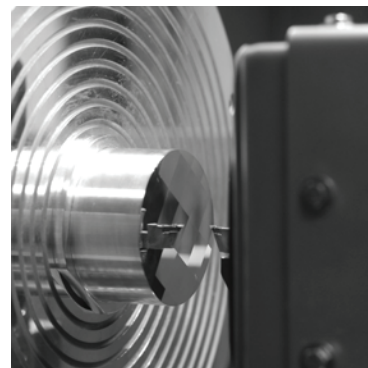


FIGURE 4. Machining of a faceted mirror using a long stroke fast tool.

Hence many different characteristics of freeform geometries can be machined using only one system. There is no need to change the machine

setup or the dynamic axis, which reduces the manual effort in advance to the manufacturing significantly.

The productivity of high accuracy freeform manufacturing using the UPC 400 is furthermore enhanced by integrated optical metrology. This enables machining and measuring of freeform surfaces inside the machine and in one-clamping setup. Thereby the work piece does not have to be moved between machining and measurement steps. This avoids waiting times and re-clamping efforts. Furthermore, the coordinate system is preserved ensuring accurate measuring. The measurement result then can be used to compensate for the measured form errors in a re-machining step [11]. The fully integrated measurement system permits to automate the correction cycle. Optionally this can be done overnight without the support of the operator. By that, both the quality and the throughput in freeform manufacturing are enhanced.

The freeform generated by the long stroke fast tool is measured by using an optical sensor, that is mounted next to the fast tool on the X-slide. It scans in a spiral path over the freeform surface (see FIGURE 5). During the measurement process the Z-axis carries out the dynamic motion to follow the freeform curvature. Therefore the Z-axis of the machine is optimized for dynamic motion. Beside metrology purposes it can be used for slow tool machining as well.

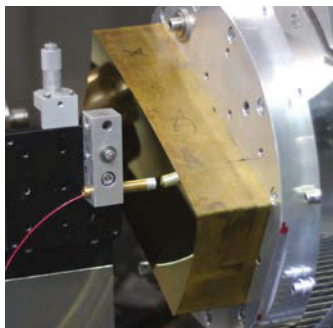


FIGURE 5. Optical measurement of on-axis machined mold insert for a HUD combiner.

Thus, the UPC 400 offers both techniques to machine freeform surfaces by ultra-precision diamond turning. Therefore the machine is the ideal base to demonstrate the advantages of a long stroke fast tool compared to slow tool machining. Nevertheless the dynamic properties of the long stroke fast tool can be expected higher, which is demonstrated in the following.

RESULTS

The comparison of the slow and fast tool freeform machining capabilities of the UPC 400 is demonstrated by off-axis machining of two spherical test cavities as shown in FIGURE 6. This provides an ideal test surface as it requires the full freeform capabilities of the machine but produces a surface that can be easily characterized to a high level of accuracy using a standard laser interferometer.

The test geometry consists of two opposing spheres located at a radius of 45 mm. The radius of each sphere is 100 mm and has a sagittal depth of 4 mm.



FIGURE 6. Off-axis spheres machined into an aluminum plate (EN AW-6082) using the UPC 400.

First, the part has been manufactured using the slow tool functionality of the UPC 400. The finish cut takes 105 min. The resulting spherical form error is better than $\pm 0.17 \mu\text{m}$ (PV) and $0.063 \mu\text{m}$ (RMS). FIGURE 7 shows the result of the interferometer measurement of one sphere. This very good result has been achieved without re-machining to compensate for the measured form deviations.

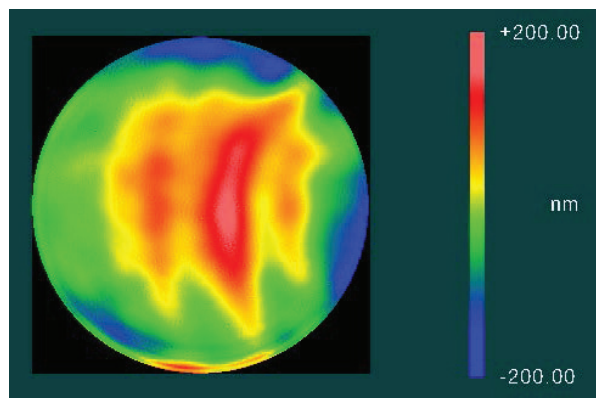


FIGURE 7. Result of slow tool machining.

In a second step the part was machined by the fast tool system on the UPC 400. Due to the higher dynamic properties the rotational speed could be increased. Using the same tool and process parameter this reduced the machining time to 35 min, which is a third of the original machining time. FIGURE 8 shows the result of the form measurement. The measured form deviation is $\pm 0.19 \mu\text{m}$ (PV) and $0.070 \mu\text{m}$ (RMS). Again this is the result without any compensation.

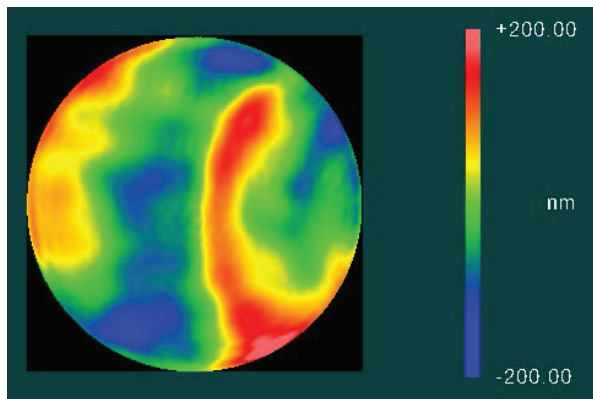


FIGURE 8. Result of fast tool machining.

The results demonstrate that the off-axis spheres can be machined by the fast tool system to the same level of form accuracy and a significantly shorter process time.

CONCLUSION

The fast tool on the UPC 400 enables very productive and flexible machining for a wide range of freeform surfaces. Its extraordinary stroke and dynamic properties allow for significantly reduced process times compared to slow tool machining while maintaining the same level of accuracy.

In addition this reduces the impact of external influences allowing for a more accurate, robust, and repeatable manufacturing process. The combination of powerful data handling capabilities, diamond turning, milling, and metrology make the UPC 400 an ideal platform for fabricating freeform surfaces.

REFERENCES

- [1] Scheiding S; et al. Ultra-precisely manufactured mirror assemblies with well-defined reference structures. Proceedings of SPIE 7739-08. 2010.
- [2] Brecher C (Ed.). Mikrostrukturierte, lokal-funktionale Oberflächen für hocheffiziente

Beleuchtungssysteme. Apprimus. Aachen. 2012.

- [3] Dick L. High Precision Freeform Polymer Optics. Optik & Photonik. Wiley. Weinheim. 2012.
- [4] Brecher C (Ed.). Machine and Process Development for the Robust Machining of Microstructures on Free-Form Surfaces for Hybrid Optics. Apprimus. Aachen. 2009.
- [5] Patterson SR, Magrab EB. Design and testing of a fast tool servo for diamond turning. Precision Engineering. Vol. 7/3. Elsevier. 1985. 123-128.
- [6] Moriwaki T, Tateiwa S, Kagawa T. Development of micro tool positioning system for precision machining. Proceedings of the International Conference on Advanced Mechatronics. JSME. Tokyo. 1989. 147-152.
- [7] Pyra M. Nicht-rotationssymmetrische Laserspiegel. Dissertation. RWTH Aachen. 1994.
- [8] Ludwick SJ, et al. A Rotary Arm Based Turning Machine for Ophthalmic Lenses. Proceedings of the ASPE Annual Conference. St. Louis. 1998.
- [9] Özmeral H. Elektromagnetisches Fast Tool Servo System für den Einsatz in der Ultrapräzisionstechnik. Dissertation. RWTH Aachen. 2000.
- [10] Niehaus F. Herstellung von diskontinuierlichen Mikrostrukturen durch Ultrapräzisionszerspanung. Dissertation. RWTH Aachen. 2012.
- [11] Niehaus F, Huttenhuis S, Pisarski A. Corrective Machining of Freeform Surfaces Using the UPC 400. Proceeding of the ASPE Annual Conference. Boston. 2014.

THE LEGO WATT BALANCE: A SIMPLE INSTRUMENT TO DEMONSTRATE PRECISION METROLOGY

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ABSTRACT

A watt balance is an electromagnetic force balancing instrument to realize the unit of mass at the kilogram level. A table-top sized watt balance constructed from LEGO bricks is a cost effective version that is capable of measuring gram-level masses to 1% relative uncertainty. It is a great demonstration tool for educational outreach purposes and, most importantly, bridges metrology to entertainment. We briefly describe the construction, measurement procedures, and data collected by the LEGO watt balance.

INTRODUCTION

A redefinition of the International System of Units, the SI, is impending and may occur as early as 2018. Specifically, in the context of mass metrology, the base unit kilogram will be redefined via a fixed fundamental constant. A watt balance is one method of accomplishing this realization; however, the apparatus can seem complex and intimidating upon first glance, even though the physics behind its operation is fairly elementary. Under the inspiration of Terry Quinn, the BIPM Director Emeritus, we have constructed a handful of LEGO watt balances to illustrate these concepts through an approachable electromechanical apparatus that serves to demonstrate some key precision metrological techniques.

TRUNCATED WATT BALANCE THEORY

For all intents and purposes, the watt balance can be oversimplified into two main components, a coil and a radial magnetic field, where their interaction reveals the end game – generating an upwards electromagnetic force equal to the downward gravitational force exerted by a mass. In an experiment, the system toggles between opposite modes of operation and can be loosely compared to the inverse relationship of a motor and a generator. In Velocity Mode (VM), the coil is moved orthogonally relative to the radial

magnetic field thus creating a voltage (like a generator). In Force Mode (FM) the same coil is injected with an electric current to produce an electromagnetic force (like a motor). The watt balance equations are

$$V = BLv$$

and

$$mg = BLI$$

for VM and FM, respectively, where V is the induced voltage, BL is the product of the magnetic field and the total length of wire in the coil, v is the velocity at which the coil moves, g is the local acceleration of gravity, I is the electrical current, and m is the test mass to be measured.

By combining equations, we cancel out BL to virtually compare electrical power to mechanical power,

$$VI = mgv.$$

Like in many cases involving precision metrology, the overarching physical theory is clear cut and understandable, but designing and building an apparatus capable of achieving such a measurement... well, therein lies the challenge.

THE LEGO WATT BALANCE DESIGN

We chose a symmetric design for the LEGO watt balance that visually conforms to a common equal-arm beam balance. Fig. 1 shows a CAD rendition of our instrument. Two identical weighing pans are suspended off each arm of the balance beam which pivots about its central T-brick (see Fig. 2). Suspended below each mass pan is a wire-wound coil immersed in a radial magnetic field.

Each magnetic field is generated by two neodymium (N48) ring magnets oriented in repulsion, spaced approximately 2 cm apart. These magnets are horizontally constrained by a

brass threaded rod and vertically constrained with hex nuts. This design allows us to adjust the distance between the magnets and the geometric center of the assembly. The threaded rod is then fixed to a small, movable, wooden base plate. Each coil hangs concentric to its corresponding magnet assembly.

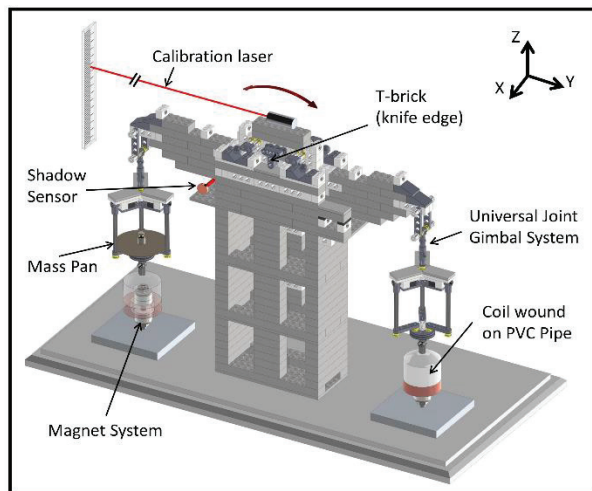


FIGURE 1. CAD model of the LEGO watt balance. The balance pivots about the T-brick at the center. Two PVC endcaps with copper windings hang from either side of the balance beam. Coil A is on the left and Coil B is on the right. A 10 gram mass sits on the Coil A mass pan and each coil is concentric to its own magnet system. Two lasers are used to calibrate and measure the linear velocity of each coil.

Each coil former was made from a small section of PVC pipe glued to an end cap. The coil was manually wound using a lathe set at a very slow rotational speed directly onto the PVC section. Every few layers of the windings were potted with epoxy. We chose to use AWG-36 wire with about 3000 windings. In our system, a current of 2.7 mA generates about 0.1 N of force. The total resistance of each coil is about 450 Ohms.

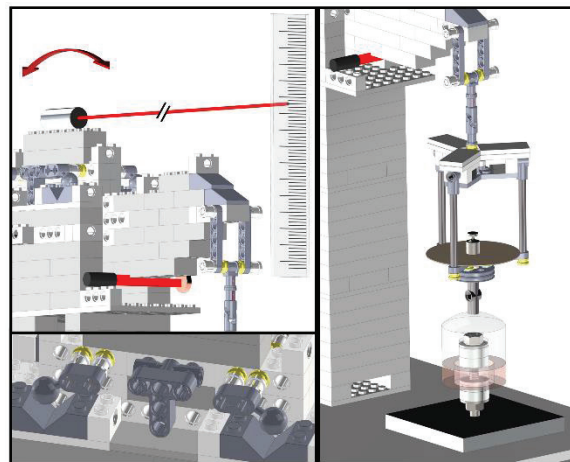


FIGURE 2. Top left: The calibration laser projects onto a ruler a few meters away. The shadow sensor detects angular motions of the balance and outputs an oscillatory voltage signature. Right: A transparent view of the PVC coil assembly shows its concentricity to the magnet assembly. A stainless steel 10 g mass sits centered on the mass pan. Bottom left: LEGO T-brick serving as the central pivot with balls and V-blocks for kinematic realignment. An identical set exists on the opposing face of the balance.

The whole balance measures approximately 43 cm x 36 cm x 10 cm and has a mass of 4 kg, including the wooden base board.

ELECTRONICS AND DATA ACQUISITION

We employ two USB devices, a U6 from Labjack and a 1002_0 from Phidget, to connect the LEGO watt balance to a laptop. The U6 is used to measure the position of the balance beam, the induced voltage, and current in each coil. The 1002_0 is a four channel analog output that is capable of producing a voltage between -10 V and +10 V. Each channel can source up to 20 mA. One channel is used for each coil.

Recently, we have verified that the NI USB-6001 will function as an alternative data acquisition device for the Labjack/Phidget combo. Though the USB-6001 has slightly lower resolution and sensitivity, the overall price of it is much cheaper than the Labjack/Phidget. Two executable software packages were created to control each set of hardware.

VELOCITY MODE

Like the real watt balance, the LEGO watt balance must undergo a series of alignments and calibrations prior to starting the experiment.

A full measurement run involves the execution of VM then subsequently FM. The purpose of VM is to calibrate and prepare the LEGO watt balance for FM by indirectly measuring a precise BL .

As stated earlier, to execute VM and measure an induced voltage in the coil, the coil must be moved vertically up and down at a known velocity. The LEGO watt balance employs a shadow sensor for measuring angular displacement. The shadow sensor consists of a line laser shining onto a photodiode. As the balance beam oscillates, it clips the line laser and casts a shadow onto the photodiode, generating a sinusoidal voltage signature.

This signal is then translated into a varying displacement through a second calibration laser mounted on top of the balance beam. The calibration laser shines onto a vertical ruler, typically spaced about 3 meters away. We servo the balance to a few arbitrary angles and linearly correlate the shadow sensor output to the laser spot's relative displacements on the wall. Finally, taking the time derivative of said displacement yields a velocity.

FORCE MODE

In FM, the same coil previously used for generating an induced voltage now receives a current to "levitate" a mass. First, a null reading is taken when the balance is empty and level. We then carefully place a mass (between 5-10 grams) onto the mass pan and the control loop adjusts the amount of current until the balance is back in its null position. The computer records the amount of current needed and translates this value into a mass reading in grams. Finally, the mass is removed and a second null reading is taken to account for drift (a typical A-B-A measurement technique).

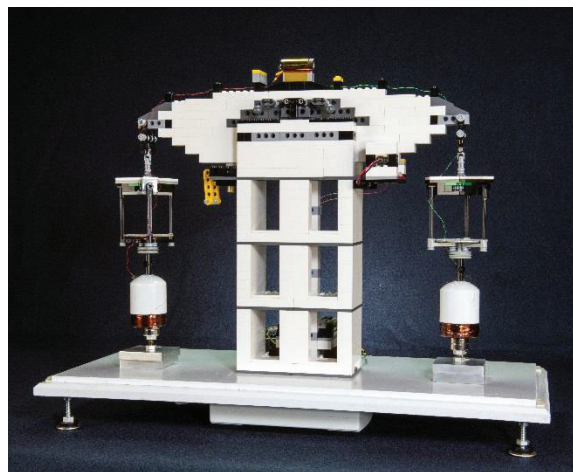


FIGURE 3. A photograph of our first prototype LEGO watt balance. Later models have slight geometric and electronic variations but the measurement techniques remain the same.

For this specific measurement campaign, we chose to operate FM in the same fashion as the real watt balance where we kept a tare mass on the far side of the balance equal to half the weight of the test mass. Two advantages of this technique include the ability to weigh larger masses without needing to increase any aspect of our electromagnet system and to reduce the effects of pivot friction. For example, by keeping a 10.0 g mass on the far side of the balance, we were able to measure a 20.2 g test mass with an electromagnet capable of generating ± 0.1 N of force. A jewelry scale with 10 mg resolution was used to determine the mass of the test mass.

DATA ANALYSIS

Averaging is a powerful metrological tool when it comes to decreasing a measurement's random uncertainty. For a relatively lower uncertainty VM measurement, we measured over 80 cycles, each period spanning about 1.5 seconds.

Fig. 4 shows the voltage and velocity data taken over 120 seconds. The slope of the best fit line in the top graph indicates the indirectly measured BL of one cycle. For this set, the average BL over 80 cycles was determined to be 36.65 Tm and the relative standard deviation was 0.2% -- a small but still worthy contributor of uncertainty for our sub-1% goal.

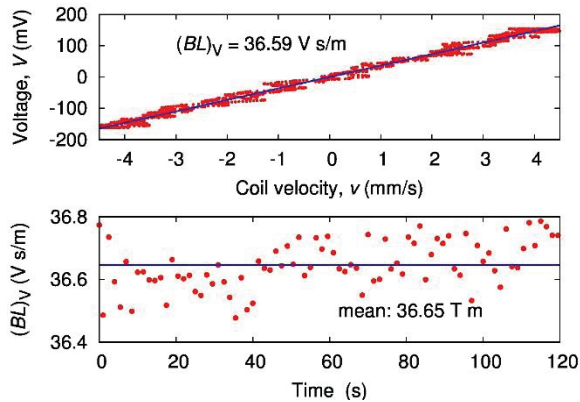


FIGURE 4. Top: The velocities and voltages were measured for one period of 1.5 s. The slope of the best fit line represents the BL associated with this cycle. Bottom: Averaging all 80 determinations yield a mean BL of 36.65 T m.

Finally, Fig. 5 shows the measurement sequence with each determination lasting 30 seconds. The top graph indicates the amplitude of each perturbation as the 20.2 g mass was placed and removed from the mass pan with steady tweezers. The bottom graph shows the amount of current necessary to level the balance when toggling between mass-loaded and mass-unloaded states.

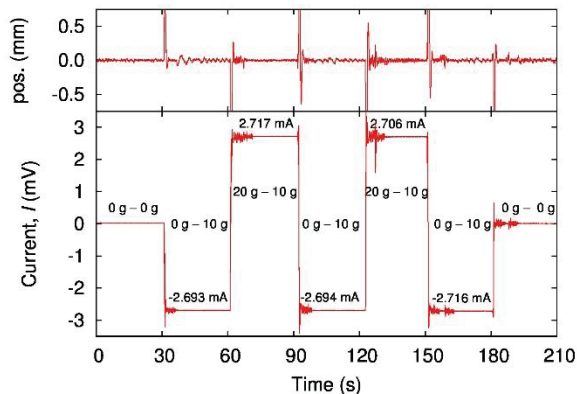


FIGURE 2. Top: The perturbation amplitude of each time the test mass is loaded and unloaded from the mass pan. Bottom: The current values necessary for holding the null position for each 20.2 g test mass loading and unloading.

The sum of the average currents from three mass-unloaded and two mass-loaded readings equals 5.413 mA. Using the $BL = 36.65 \text{ T m}$ from VM and $g = 9.81 \text{ m/s}^2$,

$$m = \frac{BLI}{g} = \frac{(36.65)(.005413)}{9.81} = 20.2 \text{ g.}$$

CONCLUSION

By carefully following the procedures and staying vigilant of proper metrological practices, an operator can measure small masses with an uncertainty below 1%. If you are interested in building your very own watt balance with LEGO bricks, we have compiled a Parts List¹ and currently have two versions of our software available upon request. Our dream is to cultivate a group of LEGO watt balance enthusiasts so that this apparatus will continually develop and evolve. Please also visit our Facebook page² and our instructional Youtube video³ for more information.

REFERENCES

- [1] Chao, L S et al. (2014). "A LEGO Watt Balance: An apparatus to demonstrate the definition of mass based on the new SI." <http://arxiv.org/abs/1412.1699>
- [2] www.facebook.com/LEGOwattbalance
- [3] <http://www.nist.gov/pml/nist-diy-watt-balance.cfm>

MICROPHONE SQUEAL AND MACHINING CHATTER

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INTRODUCTION

Self-excited vibration, or regenerative chatter, in machining has been studied for many years. The research topic has encouraged the application of mathematics, dynamics (linear and nonlinear), controls, heat transfer, tribology, and other disciplines to the field of machining science. In particular, a foundational understanding of regenerative chatter was established by the application of the Nyquist criterion from controls theory to the feedback system generated by machining processes [1]. The purpose of this paper is to demonstrate the analogy between the squeal frequency experienced when audio feedback causes instability in public address systems [2] and the chatter frequency exhibited by machining operations. A simple microphone/speaker setup is described and experimental results are quantitatively and qualitatively compared to machining stability predictions.

MACHINING STABILITY MODEL

To describe the linear stability analysis, turning is considered here. However, the approach may be extended to milling and other machining processes. When a flexible tool is used to cut away material in the form of a chip, the corresponding cutting force causes time-dependent deflections of the cutting tool. As the vibrating tool removes material, these vibrations are imprinted on the workpiece surface as a wavy profile. Figure 1 depicts an exaggerated view, where the initial impact with the workpiece surface causes the tool to begin vibrating and the oscillations in the normal direction to be copied onto the workpiece [3]. When the workpiece begins its second revolution, the vibrating tool encounters the wavy surface produced during the first revolution. Subsequently, the chip thickness at any instant depends both on the tool deflection at that time and the workpiece surface from the previous revolution. Vibration of the tool causes a variable chip thickness in the sensitive direction, y , which yields a variable cutting force since the force is proportional to the chip thickness. See Eq. 1, where F is the cutting force, K_s is the

specific cutting force, b is the chip width, h_m is the mean chip thickness, $y(t - \tau)$ is the vibration in the previous revolution, y is the current vibration, and τ is the time delay (i.e., the time for one revolution). The force is separated into the mean, F_m , and variable, F_v , components. Because the cutting force governs the current tool deflection through the structural dynamics, the system exhibits feedback.

$$F = K_s b (h_m + y(t - \tau) - y) = K_s b h_m + K_s b (y(t - \tau) - y) = F_m + F_v \quad (1)$$

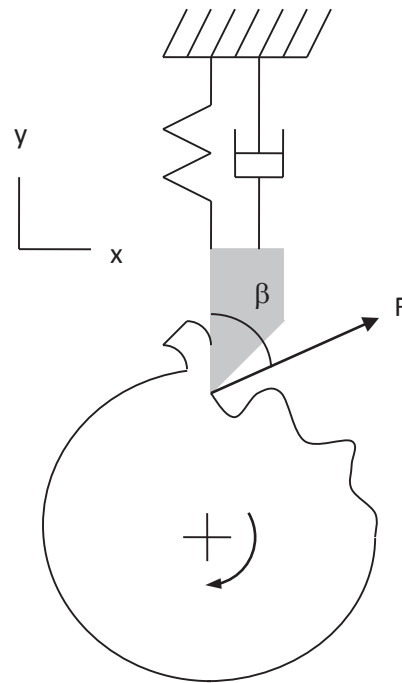


Figure 1: A flexible tool deflects due to the cutting force [3]. These deflections are imprinted on the workpiece surface and cause a variable chip thickness in the next revolution.

Equation 1 can be expressed as a block diagram as shown in Fig. 2. The y -direction projection of the variable force component (through the force angle, β) excites the structural dynamics in the y -

direction. The dynamics are described using the frequency response function, or FRF, in that direction. The time-delayed deflection, $y(t-\tau)$, is represented using the exponential function, where $\varepsilon = \tau\omega$ is the phase (in rad) between current and previous tool vibrations and ω is the frequency (rad/s).

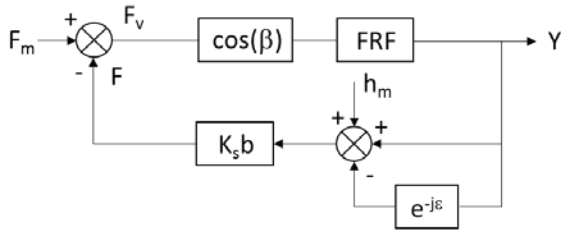


Figure 2: Block diagram description of the turning process [1]. The variable component of the cutting force excites the structure's FRF after projection into the y-direction. The corresponding frequency domain vibration, Y, is summed with the vibration from the previous revolution and mean chip thickness, and finally scaled by the product of the specific cutting force and chip width to produce the cutting force.

The Nyquist criterion is applied to Fig. 2 to identify the limit of stability, where the chip width, b, acts as the system gain. The limiting chip width, b_{lim} , is obtained when the open loop transfer function is set equal to -1 as shown in Eq. 2.

$$K_s b_{lim} FRF(1 - e^{-j\varepsilon}) = -1 \quad (2)$$

Equation 2 can be rewritten to relate b_{lim} to the real (Re) part of the FRF as shown in Eq. 3. Additional relationships for the phase and spindle speed are provided in Eqs. 4 and 5 [3].

$$b_{lim} = \frac{-1}{2K_s \cos(\beta) \text{Re}[FRF]} \quad (3)$$

$$\frac{f_c}{\Omega} = N + \frac{\varepsilon}{2\pi} \quad (4)$$

$$\varepsilon = 2\pi - 2 \tan^{-1} \left(\frac{\text{Re}[FRF]}{\text{Im}[FRF]} \right) \quad (5)$$

In Eq. 4, f_c is the chatter frequency (should it occur), Ω is the spindle speed, N is the integer number of waves of vibration imprinted on the

workpiece surface in one revolution, and $\frac{\varepsilon}{2\pi}$ is any additional fraction of a wave. Note that for units consistency in Eq. 4, if f_c is expressed in Hz, then Ω must be specified in rev/s.



Figure 3: Experimental setup. The speaker is on the left and the unidirectional microphone is on the right. The distance between them is determined using the steel scale.

EXPERIMENTAL SETUP

In public address systems when a microphone/speaker combination is used to amplify the microphone input, a similar feedback phenomenon is observed. In this case, the speaker output can re-enter the microphone after some delay defined by the time for the sound to travel from the speaker to the microphone. This generates a time-delay feedback system analogous to the machining case. An experimental setup is constructed to explore the system instability, which is exhibited as an audible squeal frequency, and demonstrate its similarity to the chatter frequency in machining.

The setup is pictured in Fig. 3, where a Radioshack Mini Amplifier/Speaker and a Switchcraft unidirectional microphone are aligned and the distance between them is varied. The distance is observed using a steel scale. The speaker volume can also adjusted to modify the system gain. The speaker volume is the analog to chip width in machining.

RESULTS

Measurements were performed to identify: 1) the system natural frequency(s); and 2) the squeal frequency, f_s , as a function of the distance between the speaker and microphone, d. The natural frequencies were determined by

introducing a known audio frequency into the microphone, measuring the speaker output, and then incrementing the frequency over the range of interest. The Fourier transform of both signals was then calculated and the ratio of the speaker output magnitude to the microphone input magnitude at the test frequency was determined (this is the sine sweep approach used in traditional modal testing). This result is displayed in Fig. 4; a modal fit is also included, where second order dynamics were assumed. Although other fits could be considered, only two clear peaks (near 2000 Hz and 4000 Hz) were evident and it was believed that the other points were near the noise floor of the measurements. The modal parameters for the two-mode fit are provided in Table 1. For these measurements, the Tone Generator application [4] for the iPhone was used as the frequency source and the Voice Memos application (in the iPhone utilities menu for a second iPhone) was used to record the speaker output (44.1 kHz sampling rate with no additional digital filtering). The .m4a sound files were then analyzed in Matlab®.

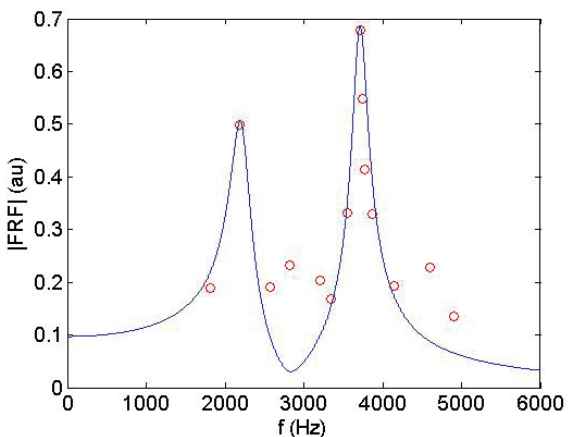


Figure 4: Ratio of the speaker output magnitude to the microphone input magnitude in arbitrary units, au (circles). A modal fit (solid line) is also shown.

Table 1: Parameters for modal fit to microphone/speaker FRF magnitude.

Mode	Natural freq. (Hz)	Stiffness (au)	Viscous damping ratio
1	2215	17	0.06
2	3713	26.2	0.028

The modal parameters in Table 1 were used to approximately represent the system dynamics as a frequency response function, FRF, and the stability limit and squeal frequency were

predicted using Eqs. 3-5. The time delay, τ , was first calculated as the inverse of the predicted spindle speed vector from Eq. 4. The time delay was then related to the distance between the speaker and microphone using Eq. 6, where v is the velocity of sound in dry air at 20 deg C (343 m/s).

$$d = v\tau \quad (6)$$

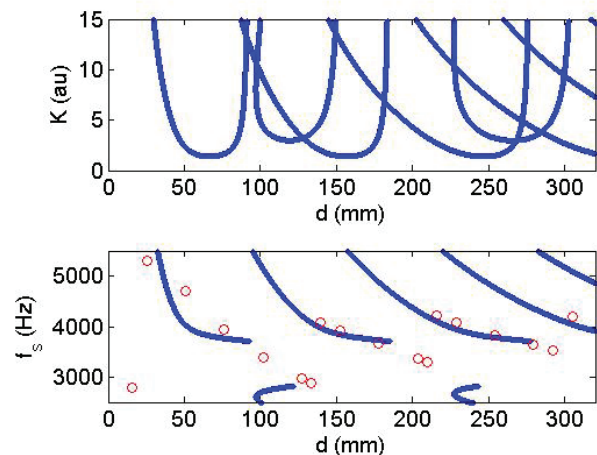


Figure 5: (Top) Stability prediction as a function of distance between the speaker and microphone, d , and the audio system gain, K (arbitrary units, au). The solid lines indicate the stability boundaries (unstable above). (Bottom) Squeal frequency, f_s , prediction as a function of d (solid line). Experimental results are shown as open circles.

Figure 5 (top panel) shows the stability limit as a function of the gain, K , and distance. Any $\{d, K\}$ combination above the stability boundaries is predicted to be unstable (i.e., squeal occurs). The vertical axis gain is the chip width in machining, but depends on the speaker volume and acoustic cavity (the volume surrounding the speaker and microphone) for the public address system. Because the FRF test results displayed in Fig. 4 were obtained with the speaker and microphone separated (to separate the source and response measurements), they do not include the inherent acoustic cavity effects. Therefore, the gain units are not provided and only qualitative comparisons between the speaker volume (gain) and stability can be completed; when applying Eq. 3, the product $K_s \cos(\beta)$ was assumed to be unity.

Figure 5 (bottom panel) shows the chatter frequency as a function of the distance between the speaker and microphone. In this case, the squeal frequency could be measured using the

Voice Memos application so a direct comparison is possible. The chatter frequency prediction from the machining stability analysis is shown as the solid line, while the squeal frequency measurements are plotted as circles.

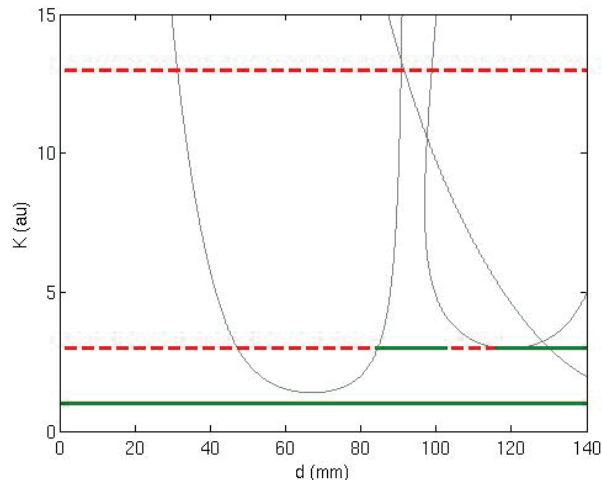


Figure 6: Stability comparison at three speaker volume (gain) values. The highest gain (top line) exhibited squeal at all locations. The dotted line indicates squeal/instability. This gain provided the squeal frequencies for Figure 5. A lower volume produced squeal only at particular distances (middle line). Stable (no squeal) distances are identified by the solid line. Below a threshold volume value, no squeal occurs (bottom line).

The squeal frequency results in Fig. 5 show good quantitative agreement with the machining analysis chatter frequencies in all cases except for the smallest distance between the speaker and microphone ($d = 15.9$ mm point). It is proposed that this unstable result is due to acoustic cavity effects and is outside the scope of this study. To provide a qualitative stability comparison, the speaker volume was varied from high to low and the regions of squeal (unstable) and no squeal (stable) were recorded. The results for three volume, or gain, values are presented in Fig. 6. Because the cavity gain was not measured, the vertical locations of the three gains (i.e., the horizontal lines in Fig. 6) were selected to approximately match the stability boundary in Fig. 6. This is equivalent to applying the machining stability predictions to a system where K_s and β are unknown. In this case it is possible to identify preferred spindle speeds (horizontal axis), but not limiting depths of cut (vertical axis). Qualitative assessment of stability is possible, but vertical scaling of the chip width is not.

While the bottom panel of Fig. 5 provides clear evidence that the squeal frequency produced by time-delay feedback in the speaker/microphone setup provides an analogy to the chatter frequency in machining, a more compelling expression of the behavior is established by audio/video from a continuous scan of the distance between the speaker and microphone. It is seen and heard that the squeal frequency periodically rises and falls as the distance is varied; the video is available online [5].

In addition, video is provided that shows both a forward (microphone moving toward the speaker) and reverse direction sweep [5]. As displayed in Fig. 8, it is observed that the dramatic shifts in squeal frequency (bifurcations) do not occur at the same spatial locations between the two directions. This indicates nonlinear behavior.

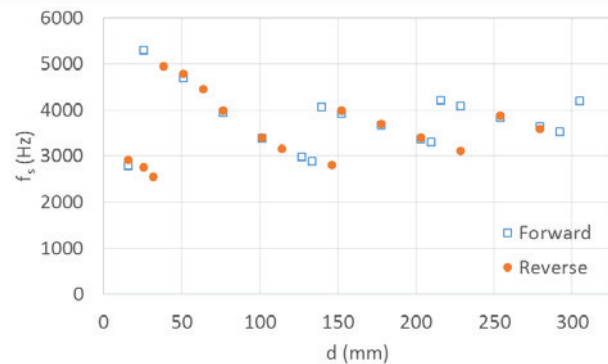


Figure 7: A discrepancy between the shifts in squeal frequency with distance is observed between the forward and reverse paths.

CONCLUSIONS

It was demonstrated that a direct analogy exists between audio feedback in public address systems and chatter in machining; both are time-delay feedback systems with stability that depends on the system gain.

REFERENCES

- [1] Tlusty J (2000) Manufacturing Processes and Equipment. Prentice Hall, Upper Saddle River, NJ, pp. 568-570.
- [2] Schroeder MR (1964) Improvement of acoustic-feedback stability by frequency shifting, The Journal of the Acoustic Society of America, 36/9:1718-1724.
- [3] Schmitz T, Smith S (2009) Machining Dynamics: Frequency Response to Improved Productivity. Springer, New York, NY.
- [4] <http://www.lifegrit.com>.
- [5] <http://coefs.uncc.edu/tschmit4/videos/>.

Carbon-graphite Gas Bearings for Turbomachinery

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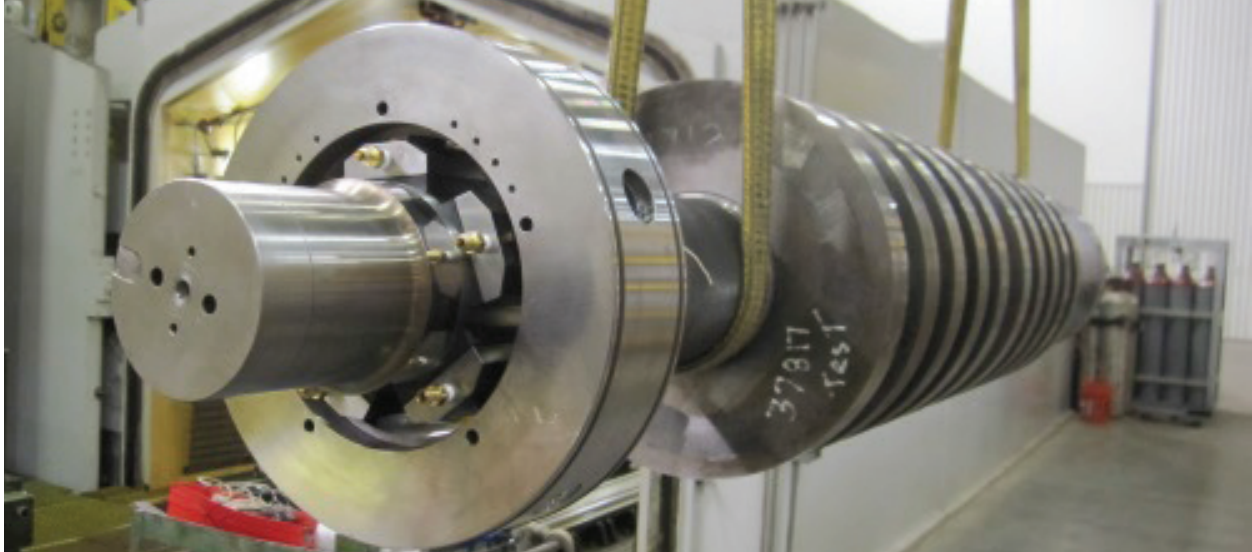


FIGURE 1. Dresser Rand test rotor mounted on New Way Gas Bearings at the TurboCare Balance Bunker in Houston, TX.

INTRODUCTION

Rotors in multi megawatt rotating equipment are almost all supported on hydrodynamic oil bearings. This has been the case for over 100 years. Kingsbury Corp., one of the inventors of tilting pad hydrodynamic bearings, is still a leader in the industry. Users of high power rotating equipment, like the power generation and the oil and gas industries, are conservative. Still there is a drive for smaller, lighter, faster, hotter, shorter, more reliable, efficient, ecological, safer and simpler equipment. It is the oil lubrication that is often the problem in achieving these goals.

Alternative bearing technologies exist; Magnetic bearings have had highly published introductions and successes although few of the high power machines use magnetic bearings. They do not extend the temperature range over oil bearings, are not compact enough for economical retrofits, and are not yet considered simple to maintain.

Aerodynamic bearings have seen wide acceptance in air handling units in commercial and military aircraft. They have also been finding use in blowers for aeration and smaller power generation turbines. Capstone, a micro-turbine

manufacturer, has a 1 MW turbine for sale now as their largest offering and it employs aerodynamic bearings. Aerodynamic bearings are compact and simple but have low load capacity and the contact at stop/start cycles becomes reliability threatening with the friction from larger rotors.

This paper explores the possibilities of externally pressurized bearings, also known as aerostatic bearings for the support of multi megawatt rotating equipment. The authors of this paper having experience with porous carbon air bearings in other industries (Semi-Conductor, Metrology, Machine Tool, Medical) employed porous carbon compensation gas bearing technology in these developments.

Aerostatic bearings provide zero friction stop and starts and have load capacity limited only by input pressure. (Authors have demonstrated stable operation in excess of 1000psi loading).

They offer the possibility of operation on process gas for canned or sealess designs at high (carbon oxidizes about 800°C) or cryogenic temperatures.

THE TEST ROTOR

Dresser Rand (DR), a leading manufacturer of large scale industrial compression technology, after being tempted with porous gas bearings by New Way, saw enough promise to challenge the authors with a particular test rotor. The rotor provided by DR had been used in the development of the Datum line of compressors. Weighing 700lbs, it is 2m long with 1.52m in between bearing centers with 100mm journals. Modeled after a 10 stage centrifugal compressor, the stages are solid discs, rather than cut impellers to increase mass and the otherwise 100mm diameter shaft is undercut to 85mm between each stage to increase flexibility. It was used as a sort of worst-case-test for bearings and dampers. The compression work being done by such a rotor is about 30MW and about 800hp is used to cool the bearing oil. Kuzdzal and Hustak [2], who worked at Dresser-Rand, conducted a study using the rotor. The authors were studying squeeze film dampers' ability to control imbalance response and rotordynamic instability.

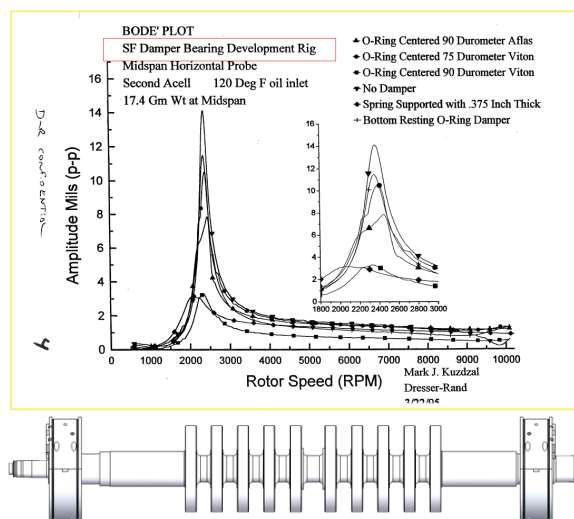


Figure 2 Using the same rotor, Kuzdzal and Hustak [1] illustrated the effectiveness of SFDs to reduce vibration amplitudes.

TEST BEARING DEVELOPMENT

The porous-media gas bearing geometry was modeled using the same 5 pad tilting-pad bearing design previously used. This was done in part to prove that direct replicability of the oil bearings with gas bearings is a possibility. The bearings were run on both nitrogen and carbon-dioxide.

The first tests conducted were aimed at characterizing bearing heat generation. The bearing cartridge containing the bearings was about 10 inches in diameter. A radial air bearing was used under the test bearing cartridge to push up with 400lbs, loading the test bearings into the spinning shaft. Two more sets of bearings and cartridges are used to hold the spinning shaft down. The test bearing cartridge is constrained in 5 degrees of freedom, has 400lbs of load yet is still able to rotate about the shaft axis

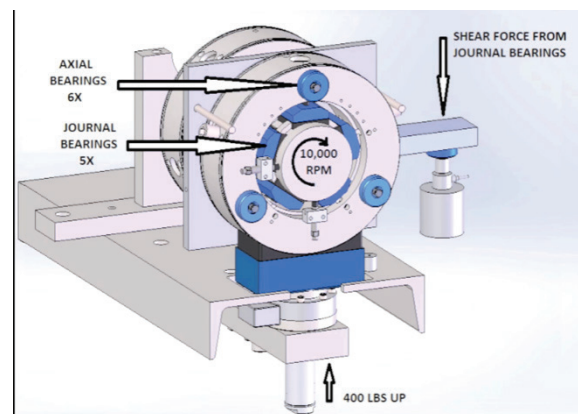


Figure 3 The test cartridge is constrained in 5° of freedom with 400lbs of radial load. A force gage can be used to directly measure bearing shear.

The authors believed they could measure the shear force between the test bearings and the spinning shaft directly by attaching an arm to a force gage, rather than looking to temperature probes and estimating losses by measuring secondary effects. Interestingly, with a 1-gram resolution force gage, results were not conclusive. Even at 14,000 RPM, the resulting drag torque was less than the resolution of the gage. The maximum power loss from bearing friction was estimated to be less than 100 watts and actual temperature probes correlated with rises less than 1°C per 15 minutes at 13,000 RPM, under load.

The temperature probes were inserted though the metal backing and down into the carbon less than 3mm from the face. Carbon has high thermal conductivity.

In the actual test with the rotor in the bunker, when supplied with 10 bar nitrogen, the rotor was able to run at 9000 rpm for 15 minutes without generating any measureable temperature rise.

While operating on CO_2 , the bearing temperatures decreased by 2°C under the same conditions.

DYNAMIC LOAD TESTING

The next test was to see how the bearings would handle vibrations from critical speeds and imbalance. Noting from the DR paper that the rotor's first critical speed was 2300 rpm, the authors fabricated imbalance weights to spin between bearing sets creating rotating forces of 200lbs, 400lbs and 800lbs at the critical speed. These imbalance forces are far beyond what would be considered appropriate by the API; however, the authors intended to induce a worst-case scenario to illustrate the usefulness of porous-media air bearings. Using eddy-current probes mounted in the pedestal, the authors were able measure motion of the whole bearing cartage in the damper and the shaft in the bearing cartridge.

As an initial benchmark, the 400lbs imbalance weight was run at 2300 rpm for 6 hours continuously without measurable change in the lateral displacements.

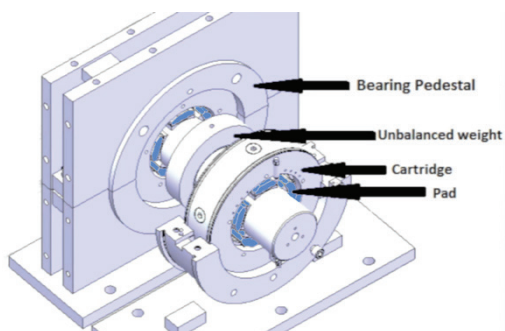


Figure 4 Unbalance weight between bearing sets

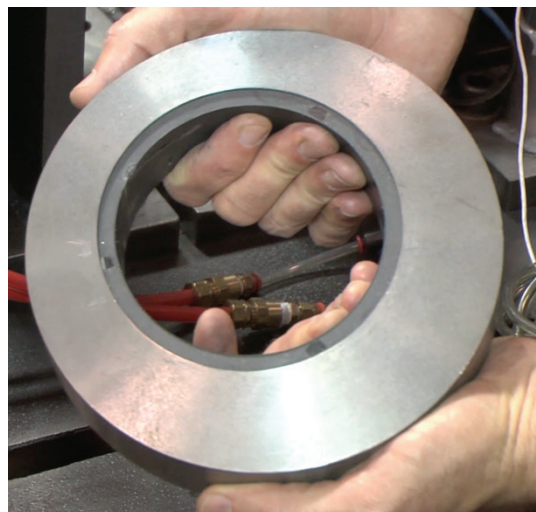


Figure 5 Actual unbalance weight spun at 2300 RPM makes 800lbs of synchronous radial load.

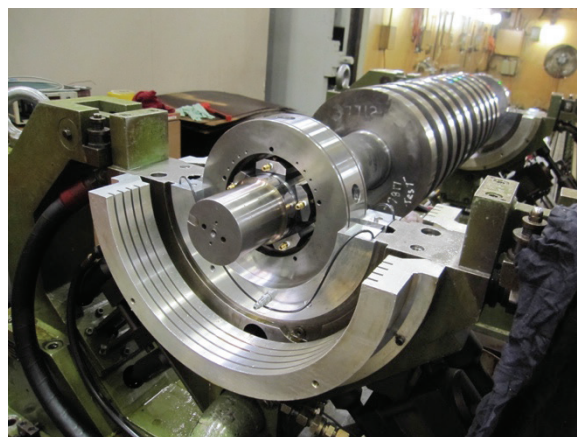


Figure 6 DR rotor, supported by New Way Air Bearings in a squeeze film oil dampers custom fit to the TurboCare bunker pedestals.

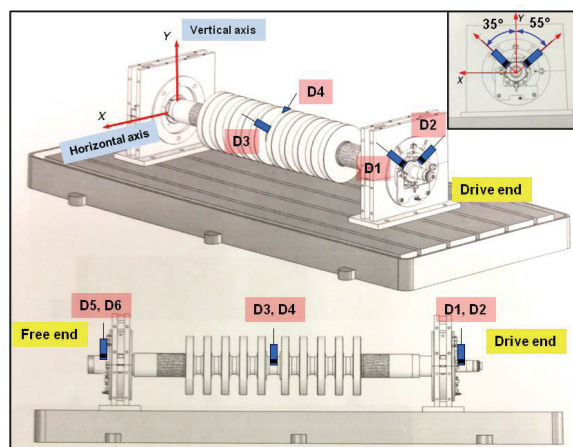


Figure 7, DR rotor on New Way Air Bearings with indicated planes of measured displacement.

FIGURE shows the response spectrum of the rotor during run-up and coast-down, measured at the bearing midspan. Note the presence of a critical speed at the mentioned 2300 rpm (38 hz).

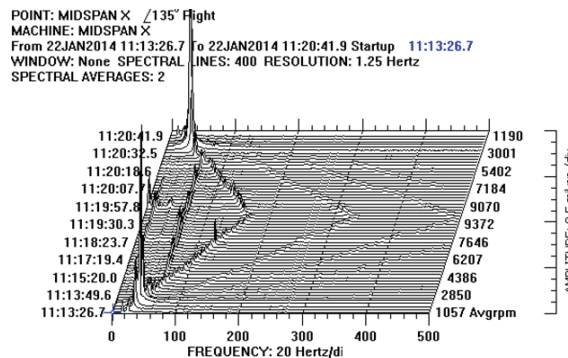


FIGURE 8. Spectrum of measured rotor response at the bearing midspan.

While cycling the actual rotor up and down on the gas bearings at TurboCare, the operators passed through the first bending critical multiple times, each time surviving an amplification factor (af) of 23 times. Additionally the authors were able to run the rotor continuously at 4 times the first critical speed (9200 RPM). The authors believe these are first-time ever achievements for gas bearings.

FIGURE 9 shows the measured rotor response at the bearing cartridges. Note the presence of another critical speed at about 9000 RPM (150 Hz).

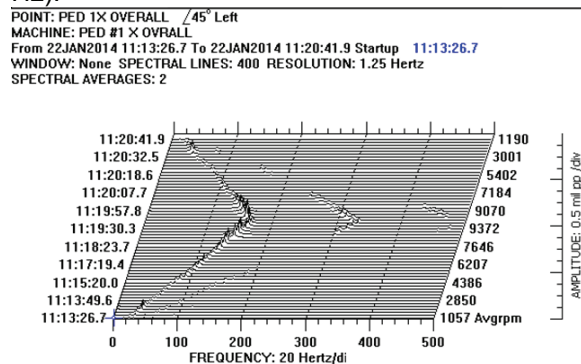


FIGURE 9. Spectrum of measured rotor response at the bearing cartridge.

Unfortunately the challenge from Dresser Rand was to spin the rotor to 10,000 RPM and at or just before the goal, the undriven bearing set failed. Eddy-current probe measurements show that the damper in the bearing cartridges experienced very little motion. The principal of

operation of a damper is that they can dissipate energy and absorb large amplitude motion. FIGURE 10 shows the relative motion of the bearing cartridge with respect to the damper. Note that there is little to no motion during both run-up and coast-down. The small motion seen is at the 1st and 2nd critical speeds (2300 and 9200 RPM).

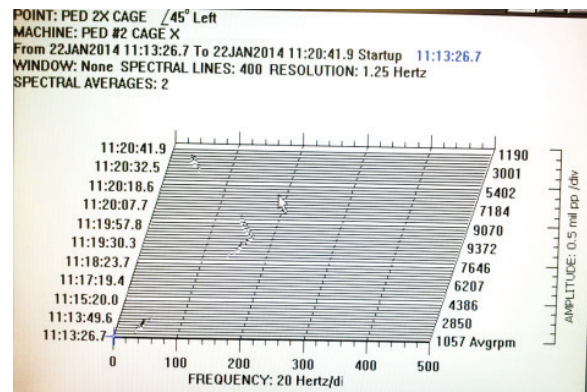


FIGURE 10. Measured relative motion between the bearing cartridge and damper.

Although the crash was rather spectacular and the bearing surfaces were both destroyed there was no metal to metal contact and the rotor journal cleaned up during regrind with less than .02mm of stock removal.

ROTORDYNAMIC MODELING

San Andres [3] assisted New Way with rotordynamic modeling, using the XLTRC² Software, developed at the Texas A&M Turbomachinery Lab.

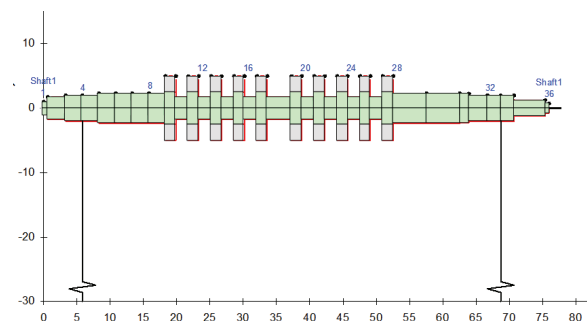


FIGURE 11. Rotor geometry modeled in XLTRC² by San Andres [6].

The code predicts critical speeds of 2600 RPM and 8800 RPM, which agrees well with test results. FIGURE 12 and FIGURE 13 show the predicted mode shapes of the rotor at the two critical speeds. Note that the damping ratio of

the 2nd critical is an order of magnitude greater than that of the 1st critical speed.

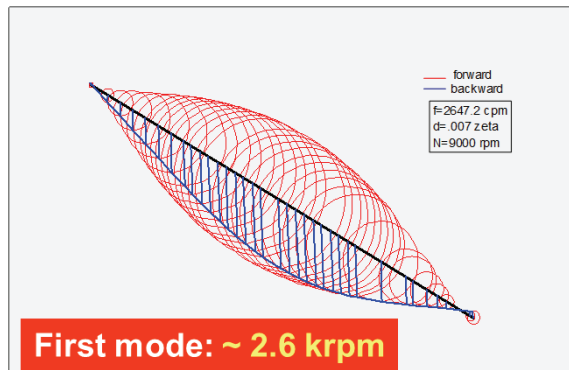


FIGURE 12. Predicted mode shape at the rotor's 1st critical speed, from San Andres [3].

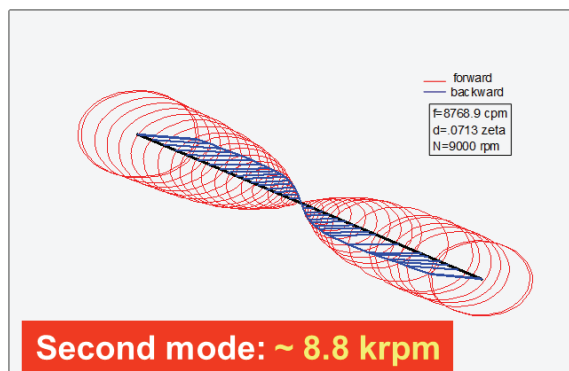


FIGURE 13. Predicted mode shape at the rotor's 2nd critical speed, from San Andres [3].

Predictions for AF also agree well with measurements taken at TurboCare. The author cannot explain why the dampers did not engage or why the measured relative motion is so small. However, a revised report by San Andres [4] shows that locked dampers would have resulted in a well-damped natural frequency at 4300 RPM, in addition to natural frequencies at 2500 and 9400 RPM. However, this revised model does not predict AF as well as initially predicted.

SUMMARY

Although the goal of 10,000 RPM was not achieved, we feel the effort was successful; passing through critical speeds with high amplification factors in relatively large flexible rotors. We will be returning to this effort with other ideas for damping that do not employ oil.

REFERENCES

- [1] G. Agrawal, "Foil Air/Gas Bearing Technology - An Overview," in International Gas Turbine and Aeroengine Congress & Exhibition, ASME, 1997.
- [2] M. Kuzdzal and J. Hustak, "Squeeze Film Damper Bearing Experimental vs. Analytical Results for Various Damper Configurations," in Proceedings of the Twenty-fifth Turbomachinery Symposium, 1996.
- [3] L. San Andres, "Proprietary Report to New Way Air Bearings," College Station, 2014.
- [4] L. San Andres, "Proprietary Report to New Way Air Bearings (2)," 2014.

COMPLIANT PIN WAFER CHUCKS FOR YIELD ENCHANCEMENT IN SEMICONDUCTOR FABRICATION

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1. INTRODUCTION

This paper describes new wafer chuck concepts to mitigate backside particle events for yield enhancement in semiconductor fabrication. Particle contamination at the interface of wafer backside and wafer chuck can lead to significant wafer non-planarity. The out-of-plane distortion can lead to overlay problems in photolithography and nanoimprint lithography, depth of focus errors in photolithography, and undesirable residual layer variations in imprint lithography, all causing yield loss.

Previously, Nimmakayala and Sreenivasan [1][2] have proposed a solution for the particle contamination issue using compliant pin chucks. The compliant pins remain rigid in normal operation. However, when a particle is present on one of the pins, the pin mechanism flexes so that the resulting out-of-plane distortion is significantly lower than with normal rigid pins. Simulations show that Nimmakayala's compliant pin chuck can potentially reduce out-of-plane distortion due to 1.5 μ m diameter particles down to 290nm and 500nm particles down to 55nm. This, however, is not sufficient for the current flatness expectations listed by the International Technology Roadmap for Semiconductors (ITRS 2013). Flatness requirement for wafers has become increasingly stringent as device feature sizes have decreased in semiconductor manufacturing. ITRS projects that wafer site flatness for 2015 would be 26nm [3].

Here, we propose three modifications to Nimmakayala's design which show significantly improved out-of-plane distortion correction. We also present experimental implementation for some of these modified designs.

2. COMPLIANT PIN CHUCK – BACKGROUND

The compliant pin chuck (CPC) [1-2] consists of a multitude of pins, each of which contains two contact lands (see Fig. 1). The pin is connected to the main chuck body using a set of secondary flexures and a flexure stem. The flexure stem provides support to the pin during normal operation and flexes when there is a

particle on one of the pins. The secondary flexures provide lateral stiffness. In addition, the secondary flexures also serve to limit out-of-plane motion of the unloaded pin when there is a particle on the other pin. There are two motion relief steps below the pin to support it in case of a catastrophic failure of the flexure elements. Compliant pins are arranged orthogonally to each other on the chuck (see Fig. 2). The compliant pin chuck is fabricated by bonding together three separately fabricated layers – the pin layer, base layer and foundation layer (see Fig. 3). The pin and base layers, which are made of silicon and contain minimum feature sizes of about 50 microns, can be fabricated using silicon micro-fabrication techniques. This design works on the principle that advanced fabrication facilities are nominally very clean, and backside particles are typically sub-micron or at the most micron scale. Furthermore, the likelihood of particles landing on two adjacent pins at the same time is extremely low.

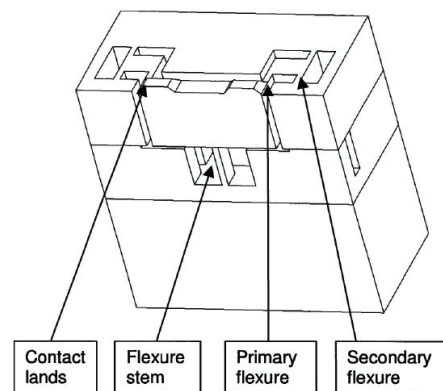


FIGURE 1. Medial section of a CPC pin [1].

3. MODIFICATION 1 – OPTIMIZATION OF COMPLIANT PIN DIMENSIONS FOR IMPROVED OUT-OF-PLANE DISTORTION CORRECTION

While better than a regular pin chuck (see Fig. 4), Nimmakayala's CPC does not satisfy current site flatness requirements. It has been found that the flexure dimensions chosen by

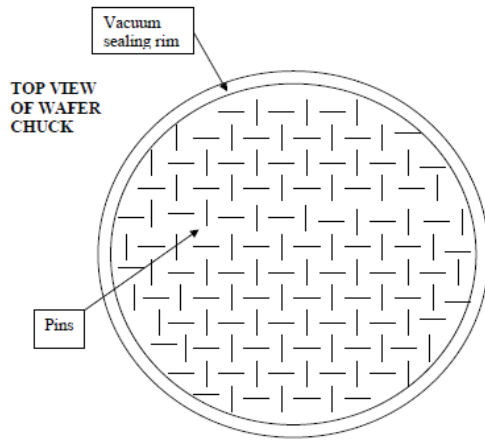


FIGURE 2. Pin arrangement on the compliant pin chuck [1].

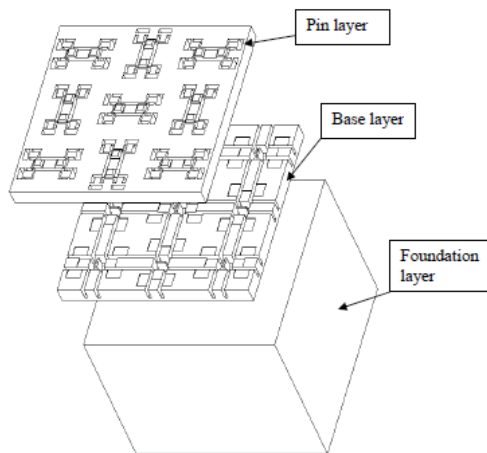


FIGURE 3. Exploded view of a part of the CPC stack [1].

Nimmakayala and Sreenivasan do not result in optimal flexure stiffness for out-of-plane distortion correction, and can be improved upon using a parameter optimization.

The optimization seeks to maximize the particle height, p_h , that a pin can accommodate for a given final out-of-plane distortion, h_{residual} (see Fig. 5). The force on a contact land due to a particle is calculated using the analytical relationship of Tejeda *et al.* [4]. The objective function also ensures that d_2 is not positive (i.e. the unloaded pin does not impinge into the wafer) and that there is no buckling failure of the pin during normal operation. Genetic algorithm is used for the optimization (see Fig. 6). Dimensional limits are imposed based on available micro-fabrication techniques.

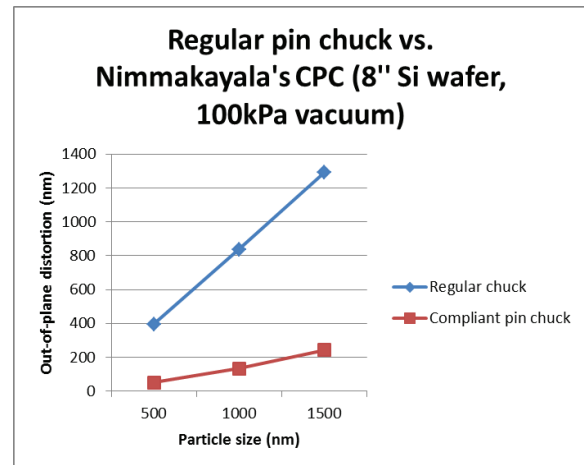


FIGURE 4. Simulation results for out-of-plane distortion in a regular pin chuck vs Nimmakayala's CPC for different particle heights.

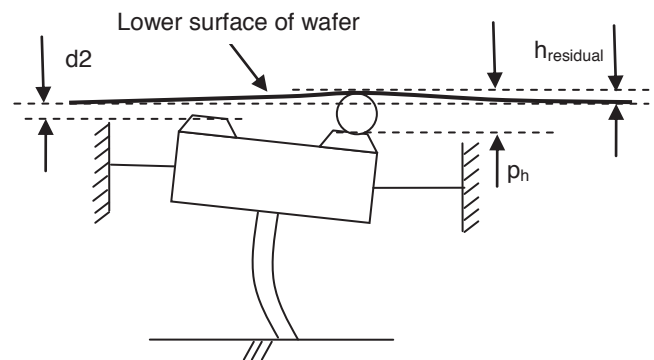


FIGURE 5. Schematic of a single CPC pin with a particle on one of the contact lands.

The optimal design is shown in Figure 7. The optimal design parameters (in mm) are as follows - $th1 = 4.1e-02$, $th2 = 0.39$, $th3 = 2.6e-02$, $th4 = 0.1$, $l1 = 0.33$, $l2 = 0.2$, $l4 = 0.3$, $stmth1 = 5e-02$, $stm1 = 1$. For a comparison of the out-of-plane distortion between this and the other designs, see Figure 10. A few things should be noted here – The current and subsequent designs address global effects induced by the backside particles. Any localized effects in the immediate vicinity of the crushed particle cannot be mitigated using these methods. Nimmakayala's original design used silicon as the wafer chuck material. For comparison, silicon has been retained in the simulation for this and the other two designs. However, SiC is the preferred material for actual wafer chucks. The described modifications can easily be

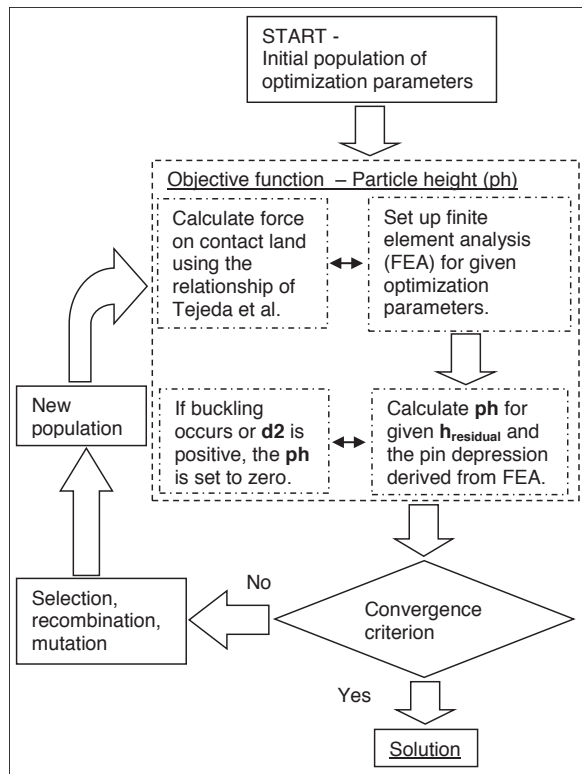


FIGURE 6. Schematic of the genetic algorithm based pin geometry optimization (for maximizing particle height, given h_{residual}).

adapted for SiC as well. Fabricating micron-scale structures in SiC is, however, more challenging than in silicon. In a later section, we will present some experimental results with a SiC based CPC that we have fabricated.

4. MODIFICATION 2 – COMPLIANT PINS WITH NOTCH FLEXURES

In this modification, secondary flexures (see Fig. 7) are changed from leaf type flexures to the more compliant notch type flexures (see Fig. 8) These reduce the stiffness of the pin mechanism in z-direction while maintaining the lateral stiffness. It should be noted that the secondary flexures serve one more function – they limit out-of-plane motion of the unloaded contact land. Since, the stiffness of notch flexures is lower than leaf type flexures, out-of-plane motion of the unloaded contact land is limited by the dual flexure stems instead. The two flexure stems limit this motion by being physically positioned under each contact land.

The design parameters (in mm) used for testing the performance of this modification (in simulation) are as follows– $th1 = 0.1$, $r1 = 5e-02$, $l1 = 0.2$, $stmd = 0.4$, $stmth2 = 2e-02$, $stml = 1$.

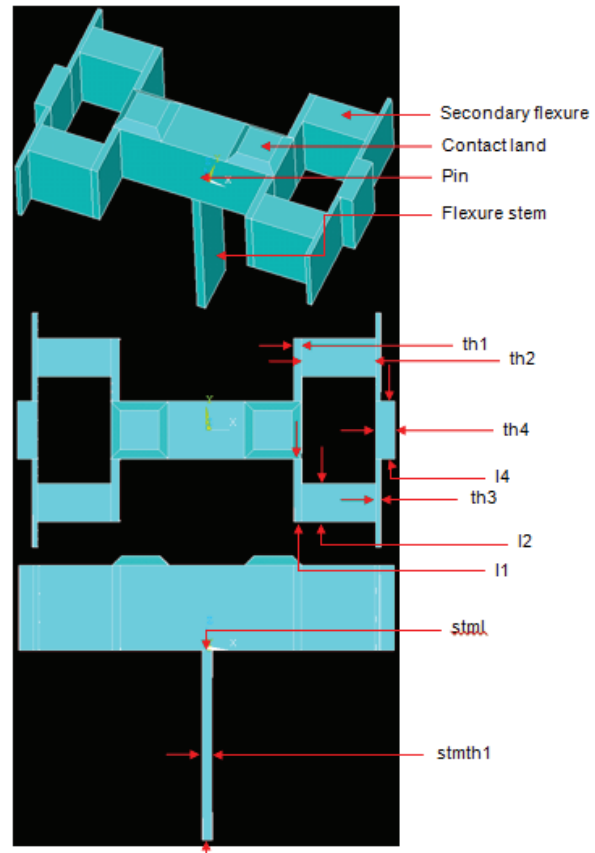


FIGURE 7. Isometric, top and side views of a single optimized pin along with the optimization parameters.

See Fig. 8 for the labelled drawings. See Fig. 10 for a comparison of the out-of-plane distortion between this and the other designs.

5. MODIFICATION 3 – COMPLIANT PINS WHICH BUCKLE FOR ASYMMETRIC LOADS

This modification, which is similar to the second one in terms of general design features, uses the principle of buckling to reduce out-of-plane distortion due to backside particles. The flexure dimensions are designed for the mechanism to buckle if (and only if) it is asymmetrically loaded, i.e. when a particle is present on one of the contact lands. Also, the mechanism is designed to buckle only when h_{residual} (see Fig. 5) is above a fixed value to prevent en masse buckling due to minor topography variations across the wafer.

The design parameters (in mm) used for testing the performance of this modification (in simulation) are as follows – $th1 = 0.1$, $r1 = 5e-02$, $l1 = 0.2$, $stmd = 0.3$, $stmth2 = 1.9e-02$, $stml = 1$, fixed h_{residual} (beyond which buckling failure

occurs) = 25 nm. See Fig. 9 for a simulated pin in buckled configuration. See Fig. 10 for a comparison of the out-of-plane distortion between this and the other designs.

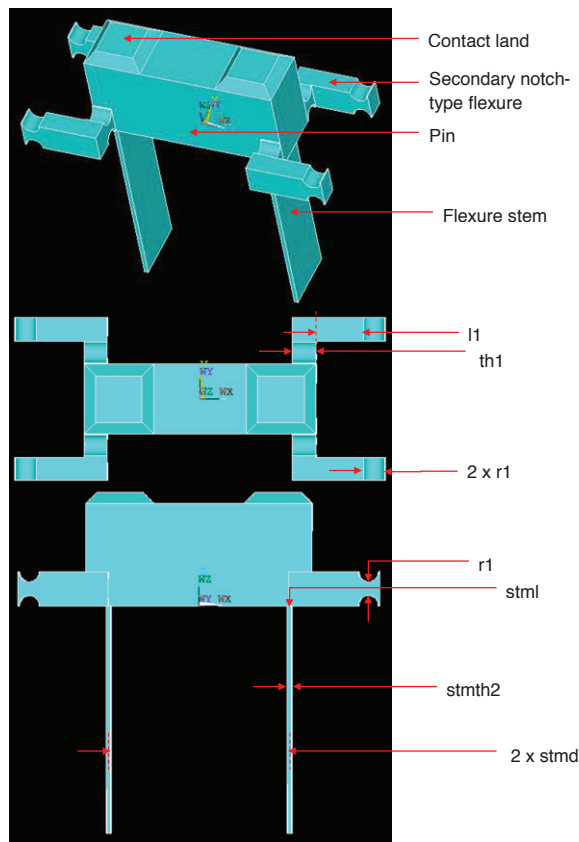


FIGURE 8. Isometric, top and side views of a compliant pin with notch flexures.

6. SiC COMPLIANT PIN CHUCKS

We have explored silicon carbide fabrication of optimized compliant pin chucks in collaboration with Nanosolutions LLC (a Connecticut based wafer chuck manufacturer), to validate the procedures of Section 3. Preliminary results are presented here in this paper. The two major challenges associated with making compliant pins out of silicon carbide are the brittle nature of silicon carbide and small dimensions of the pins. Fig. 11 shows the pin geometry and the flexures with minimum component dimensions of $50\mu\text{m}$. The fabrication has been followed by lapping to mimic the practical steps required in fabricating precision chucks for advanced lithography.

It should be noted that since the minimum component dimension was limited to $50\mu\text{m}$ in this preliminary experimentation, the optimized pin dimensions are different from the dimensions

mentioned in Section 3. With these dimensions, the simulated out-of-plane distortion for a $1\mu\text{m}$ particle is about 260nm.

7. SUMMARY

Three modifications to Nimmakayala's compliant pin chuck have been proposed which show improved out-of-plane distortion correction in simulation. Optimization techniques based on genetic algorithms have been used in this work. An actual SiC chuck fabrication has been explored to validate the optimization procedure used to come up with the design of modification 1. Future research includes optimizing the designs keeping practical SiC fabrication constraints in mind, and fabrication and process testing of a full SiC wafer chuck using programmed backside wafer particles.

8. ACKNOWLEDGEMENTS

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REFERENCES

- [1] P. K. Nimmakayala, "Development of a compliant pin chuck for minimizing the effect of backside particles on wafer planarity". Master of Science Thesis, The University of Texas at Austin. Spring, 2004.
- [2] United States Patent no. 7,259,833 B2. Sreenivasan et al. Aug. 21, 2007.
- [3] International Technology Roadmap for Semiconductors, 2013 Edition. <http://www.itrs.net/Links/2013ITRS/Home2013.htm>
- [4] R. Tejeda, R. Engelstad and E. Lovell. "Particle-induced distortion in EUVL reticles during exposure chucking". Journal of Vacuum Science and Technology B. 2002.

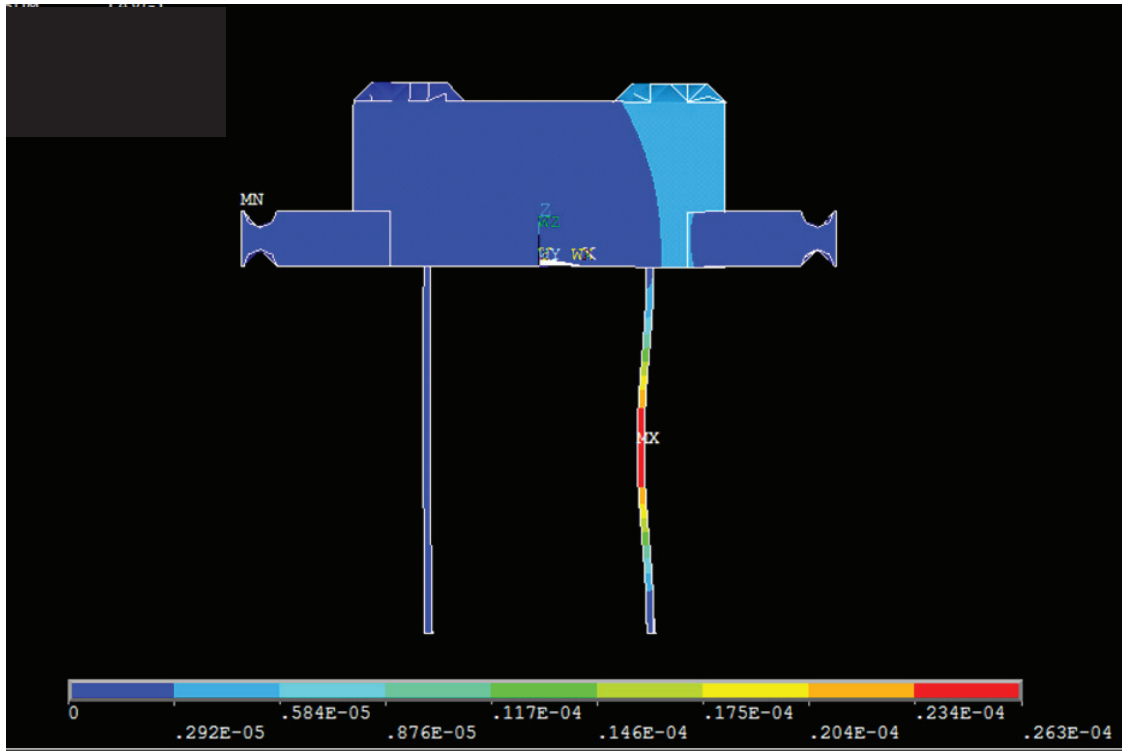


FIGURE 9. FEM simulation of a pin from Section 5 in buckled configuration. The colored regions indicate the net displacement. The values on the scale bar are in meters.

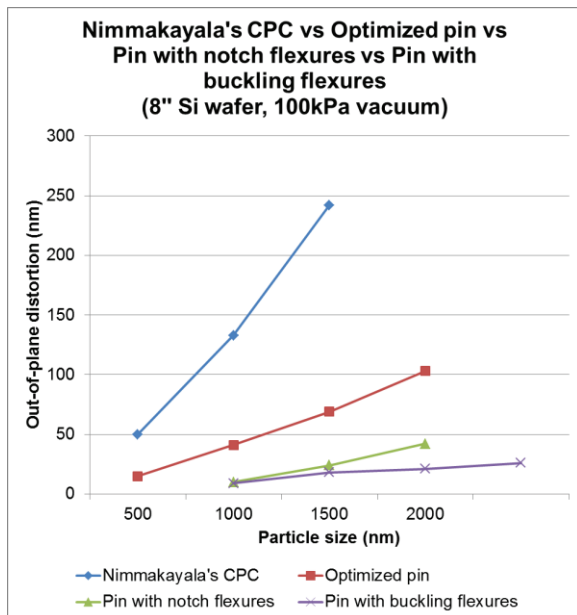


FIGURE 10. Comparison of out-of-plane distortion correction for the four discussed designs.

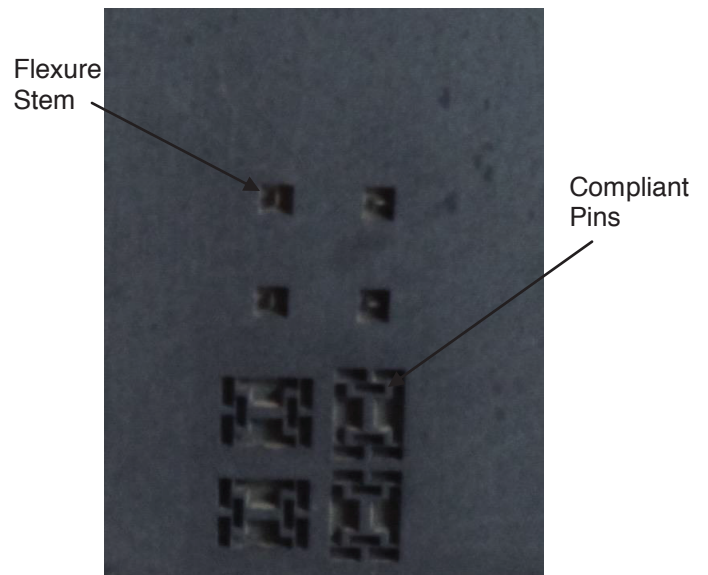


FIGURE 11. Compliant pins and flexure stems laser machined on a piece of SiC.

STUDY ON THE INFLUENCE FACTORS OF ERROR AVERAGING IN HYDROSTATIC LEAD SCREW

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INTRODUCTION

The hydrostatic lead screw has high motion accuracy due to the error averaging effect [1-3] of pressured oil film, and hence have varies application in precision or ultra-precision machine tools. Some studies have conducted on the error averaging of hydrostatic screw. Zhang found that the motion errors caused by the pitch errors of the lead screw have additivity characteristics and the pitch errors of the nut have little influence on the motion errors [4]. Sayed and Khatan optimized the traditional projection model of the screw nut pair, and studied the equivalent model of the external oil supply hydrostatic screw nut pair. This research provided theoretical guidance for the design of the structure size parameters of the hydrostatic screw nut [5]. Bassani.R proposed a hydrostatic screw nut, which is capable of self adjustment. But the structure is very complex and the machining processes are very difficultly [6]. These studies are mostly focus on theory and experiment, in order to achieve a wide range of hydrostatic screw in the ultra precision machine tool, more further researches need to be conducted. This paper investigates the influence factors of the error averaging in hydrostatic screw. The trapezoidal screw thread error model and the calculation model for hydrostatic lead screw are established by using the hydrostatic lubrication theory and the geometric model. This paper investigates the influence factors of the error averaging in hydrostatic screw. The trapezoidal screw thread error model and the calculation model for hydrostatic lead screw are established by using the hydrostatic lubrication theory and the geometric model.

THE MATHEMATICAL MODEL OF SPIRAL ERROR IN HYDROSTATIC LEAD SCREW

The schematic of the hydrostatic screw nut pair is shown in Figure 1(a), and its parameters are shown in table 1.

As shown in Figure 1(b), X_{i-j} ($i = 1, 2, 3, 4$; $j = 1, 2$) is defined as the difference between the actual

position and the ideal position on the part i of the screw thread. It indicates that the actual position is smaller than the ideal position as $x_{i-j} < 0$. Where $j=1$ represents the helical error located at the right side of the screw. And $j=2$ represents that the error located at the left side. The screw thread of the trapezoidal screw were expressed with x_{i-2} and x_{i-1} .

TABLE 1. The parameters of hydrostatic screw nut pair .

Parameters	Value
inside diameter	37 mm
outside diameter	53 mm
inside diameter	36 mm
outside diameter	52 mm
thread pitch	10 mm
spiral groove width	5 mm
spiral tooth thickness	5 mm
oil film thickness	20 μm

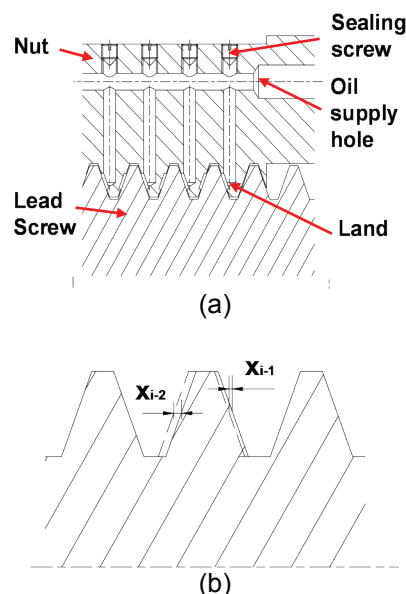


FIGURE 1. Hydrostatic Lead screw schematic:(a) the structure of the lead screw nut;(b) trapezoidal lead screw thread tooth profile.

The w_i is spiral groove width error. In the same way, $w_i < 0$ indicates the actual measurement of the spiral groove is lesser than the ideal width of the groove. Helical error of two helices can be expressed by w_i and x_{i-1} , that is:

$$x_{i-2} = -w_i - x_{i+4-1} \quad (1)$$

Similarly, the same method is used to establish the error model of hydrostatic nut helical tooth. It is represented with x_{i-1} , w_i , and can be expressed as follows:

$$w_{i-2} = -w_i - x_{i+4-1} \quad (2)$$

ERROR CALCULATION MODEL IN TRAPEZOID HYDROSTATIC SCREW NUT PAIR

Assuming that the oil film thickness of each part of the spiral surface is uniform, which is equal to the value at the central point. The oil film is considered as a rigid body, and without considering the compression of lubricating oil and the error of spiral tooth profile half angle. In the calculation formula of the bearing capacity of the helical surface equivalent model, it is assumed that the supply pressure is 4 MPa. The oil dynamic viscosity is 0.065 Pa·s. An annular slit fixed restrictor is used for pressure regulation [1-3]. The bearing capacity of the equivalent region is calculated, and the results were shown in table 2.

TABLE 2. Load bearing capacity of the equivalent region.

Oil film thickness $h/\mu\text{m}$	Carrying capacity L/N
10	684.4
12	658.0
15	605.5
18	541.0
21	471.0
24	402.0
27	338.0
30	282.1
33	235.0
35	208.0

The relationship between oil film thickness and bearing capacity is shown in Figure 2. According to the nonlinear least-square fitting method, the function of the polynomial can be derived, expressed as, (where L 's unit is N and h 's is mm)

$$L(h) = -709572473.9h^4 + 89038400.1h^3 - 3681748.1h^2 + 39802.4h + 572.15 \quad (3)$$

In order to guarantee the normal operation of the screw nut pair, 0.8 times of the maximum load in the fitting curve was used, expressed as,

$$L_w = 0.8 n_n \times 4 L_{\max} = 3.2 n_n L_{\max} \quad (4)$$

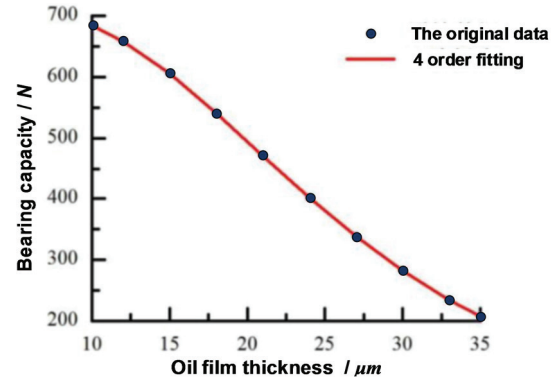


FIGURE 2. Theoretical calculation results

It is assumed that the first part of the nut is matched with the part j of the lead screw when lead screw operated, as shown in Figure 3. Load L applied on the trapezoidal nut, a displacement Δx_j yield on the left side, which lead to oil film thickness decreased Δx_j on the right side. Considering the screw thread error of the lead screw and nut, the formula for calculating the oil film thickness on the left and right sides of the helical surface of the part i can be derived, as shown in Eq.(5) and (6).

$$h_{i-1} = h_0 - (\Delta x_j + x_{i+j-1-2} + x_{i-1}) \cos \varphi \cos \gamma \quad (5)$$

$$h_{i-2} = h_0 + (\Delta x_j - x_{i+j-3-1} + x_{i-2}) \cos \varphi \cos \gamma \quad (6)$$

Where, φ : Spiral angle; γ : Tooth profile half angle; h_{i-1} : The oil film thickness of the left side of the spiral surface of the trapezoid hip screw nut i ; h_{i-2} : The oil film thickness of the right side of the spiral surface of the trapezoid screw nut i .

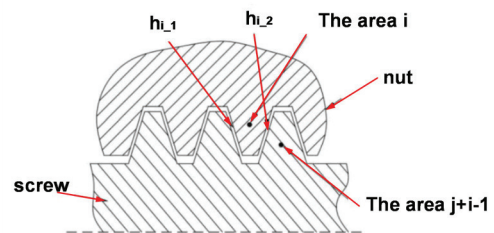


FIGURE 3. Screw nut with schematic diagram

The balanced equation for nut is given as,

$$L_w = \sum_{i=1}^{4n_n} f(h_{i-1}) - \sum_{i=1}^{4n_n} f(h_{i-2}) \quad (7)$$

According to the above mentioned descriptions, the displacement Δx_j of the load L_w can be calculated. The motion error is:

$$E_j = \Delta x_j - \Delta x_1 (j=1, 2, \dots) \quad (8)$$

In the same way, the motion error e_j ($j=1, 2, \dots$) of the trapezoidal screw nut pair can be obtained.

INFLUENCE FACTORS ANALYSIS OF ERROR AVERAGING EFFECT

Figure 4 is the measurement data of trapezoidal hydrostatic guide screw helical error. The original helical error is $3.66\text{ }\mu\text{m}$. Figure 5 is a set of screw spiral groove width and its corresponding error. The motion error results of the trapezoidal nut are shown in Figure 6. When the trapezoidal nut effective turns n_n is 2, 4, 6, and 8 respectively, the maximum motion error of trapezoidal hydrostatic guide screw on the total length is $2.56\text{ }\mu\text{m}$, $2.01\text{ }\mu\text{m}$, $1.70\text{ }\mu\text{m}$ and $1.52\text{ }\mu\text{m}$ correspondingly. With the turns n_n increased error averaging effect is more obvious. Since the nut is difficult to manufacture, the effective number of the nut turns is 4.

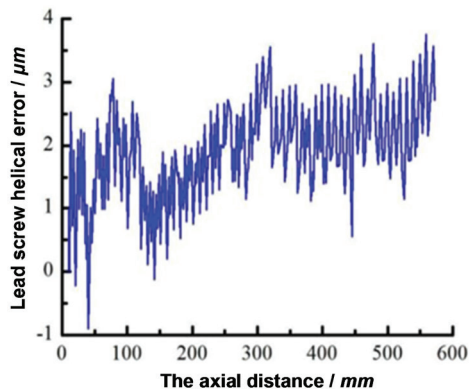


FIGURE 4. Lead screw helical error

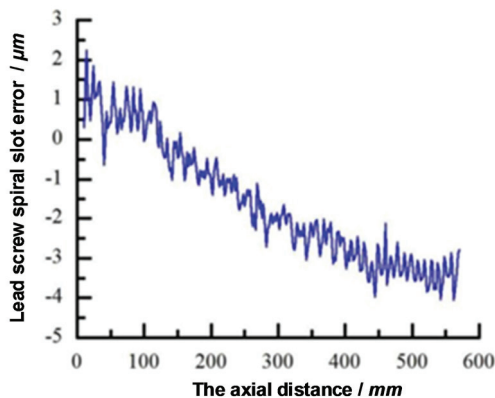


FIGURE 5. Lead screw spiral slot error

Before the pads machined, nut helical error was measured, as shown in Figure 7. When effective nut turns is 4, the influence of helical error on motion error was calculated, as shown in Figure 8. Coincidence degree is very high, considering and not considering nut helical error, which means that the nut helical error almost has no impact on motion error.

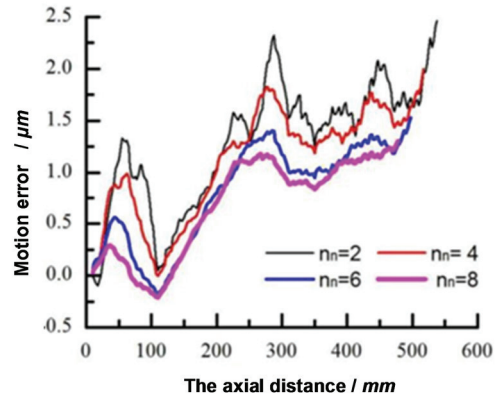


FIGURE 6. Relationship between motion error and axial distance under different n_n

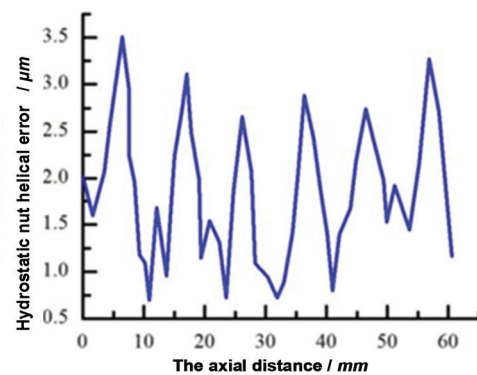


FIGURE 7. Hydrostatic nut helical error

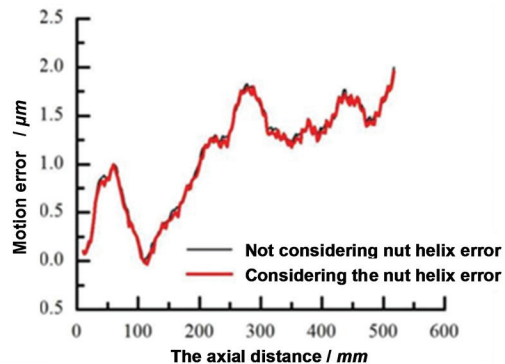


FIGURE 8. Influence diagram of nut line error on the error averaging ratio

Motion error was shown in Figure 9 when screw spiral groove width error considered and not considered. Coincidence degree is not high when considering and not considering screw spiral groove width, two motion errors are $1.67\text{ }\mu\text{m}$, $2.01\text{ }\mu\text{m}$ respectively.

When effective nut turns is 4, motion errors are calculated under different load, the results are shown in Figure 10. In Figure 10(a), it is obvious that the 3 motion error curves are almost

completely overlap when load are 0.4, 0.6, 0.8 times of maximum value. In Figure 10(b), 3 motion error values in axial position are smaller than $0.05 \mu\text{m}$. Therefore it can be concluded that influence of load size on trapezoidal hydrostatic guide screw nut pair error averaging ratio is very small, and the influence can be neglected.

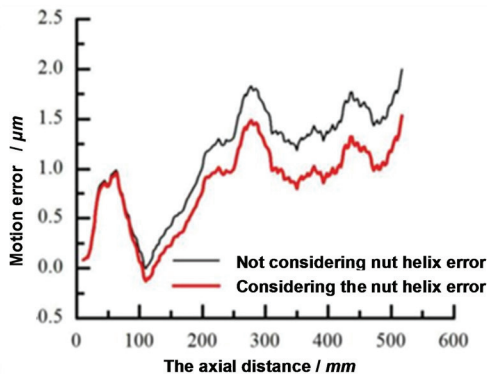


FIGURE 9. Influence of screw spiral groove width error on error averaging ratio

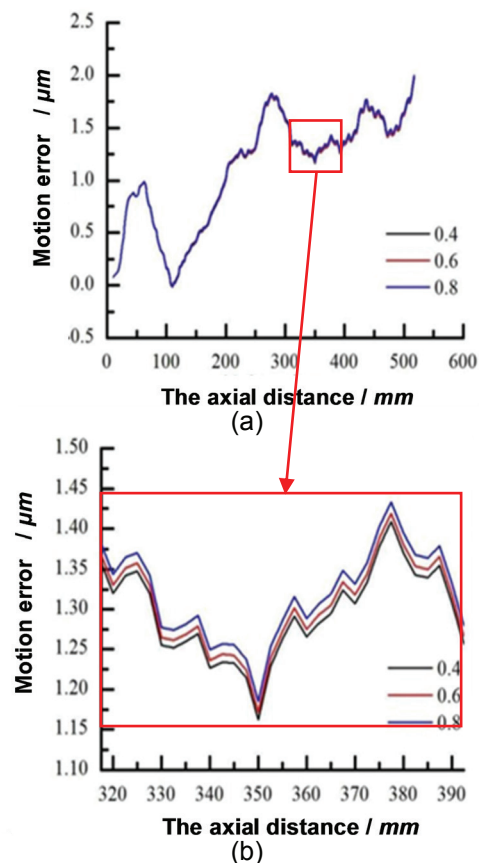


FIGURE 10. The influence of load on averaging error ratio: (a) Different load; (b) Local amplification

CONCLUSION

(1) Based on the hydrostatic lubrication theory and the geometric model of the hydrostatic screw nut, the error model of the screw thread of the trapezoidal thread and the error calculation model of the hydrostatic screw are established,

(2) Research results: The effective turns of the nut has greatest impact on the error averaging ratio. The error averaging ratio is significant as effective turns increased, and the optimal effective nut turns is 4. The effect of the nut screw thread error on motion error is small and can be neglected. Since the motion deviation is smaller as screw groove error was considered. The error of the screw groove has a significant effect of the averaging ratio on the nut pair.

(3) The research results are helpful to optimize the new structure and guide the precision design of the hydrostatic screw nut pair.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] Xue F, Zhao WH, Chen YL, Wang ZW. Research on Error Averaging Effect of Hydrostatic Guideways. Precision Engineering, 2012; 36(1), pp. 84-90.
- [2] Wang ZW, Zhao WH, Chen YL, Lu BH. Prediction of the effect of speed on motion errors in hydrostatic guideways. International Journal of Machine Tools and Manufacture, 2013; 64, pp.78-84.
- [3] Zha J, Lv D, Jia Q, Chen YL. Motion straightness of hydrostatic guideways considering the ratio of pad center spacing to guide rail profile error wavelength. International Journal of Advanced Manufacturing Technology, doi: 10.1007/s00170-015-7515-2.
- [4] Zhang YT, Lu CH, Pan W, Chen SJ, Liu ZC. Averaging Effect on Pitch Errors in Hydrostatic Lead Screws with Continuous Helical Recesses. Journal of Tribology, doi:10.1115/1.4031463.
- [5] Sayed HR, Khatan HA. A suggested new profile for externally pressurized power screws. Wear, 1974; 31 (1): 141-156.
- [6] Bassani R. The self-regulated hydrostatic screw and nut. Tribology, 1979; 185-190.

APPLYING PRECISION ENGINEERING PRINCIPLES TO HIGH VOLUME PRODUCT DEVELOPMENT

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INTRODUCTION

The term precision engineering is often mistakenly thought to apply only to low volume products, such as semiconductor wafer steppers or PCB pick and place machines. However precision implies a level of dimensional tolerance applied to any product, even high volume production. As the number of units manufactured per year increases, the ability to cost effectively maintain high levels of precision gets more difficult. For the purposes of this paper, the term "high volume" will apply to products produced in the range from hundreds to thousands of units per year. Typical industries that have production in this volume are medical, telecommunications, and industrial process management. These industries typically have business-to-business sales and have high enough margins to absorb the costs associated with manufacturing and assembly of high precision products. This paper will explore how principles of precision engineering can be applied to high volume product development.

COMPARISON OF LOW AND HIGH VOLUME PRODUCTION CONSTRAINTS

Whether the product is a one-off custom machine or a family of telecommunications products, such as server racks, precision engineering principles still apply. Two basic principles of precision engineering are needed in order to ensure the product requirements are met.

- Manage constraints - ensure that over-constraint is avoided and preventing assembled-in strain that can cause products to shift over time (example: bolting a stiff sub-assembly into a product with more than 3 mounting points)
- Deterministic assembly - Using features to guide and locate parts relative to one another so that positional variation is

minimized. This is key with high volume products so as to reduce the time required to assemble them.

A high level comparison of a low volume product versus a high volume product is shown in Table 1 (page 5). This shows how cost, performance, manufacturing and assembly processes are interrelated for particular production volumes.

While there is a definite contrast in the way low vs. high volume products are produced, the same design and assembly principles can be applied to both groups as discussed below.

CHALLENGES IN HIGH VOLUME PRODUCT DEVELOPMENT

Aside from the obvious challenge of producing high volume products at a relatively low cost, these products are also more constrained by manufacturing processes. In order to manufacture at high volumes, fabrication processes, such as stamping, molding, casting, and sheetmetal forming are typically employed as those processes can be automated. However, certain additional processes, such as machining, could be applicable to high volume products, but the decision to employ those processes will depend on material, part complexity, required tolerances, and component size.

Furthermore, with low volume production there is always the option to "move up" to a higher precision production method to improve the tolerance / precision of the produced part. For example, machining a part will result in a certain tolerance. If that is not sufficient, the next method may be to grind the surfaces. Following that process may be lapping surfaces. High volume part production is more tightly constrained to work within the typical manufacturing tolerances for that particular production method.

Once parts are fabricated they must be assembled into the final product. There is again a division between low and high volume product assembly methods. Both rely on human labor, but the level of interaction and adjustments will be different. A low volume product will rely more on manual adjustment and the use of external metrology tools for alignment. A high volume product must be assembled as quickly as possible while still maintaining the level of precision required for its intended use. Consequently, this requirement limits the amount of time and labor that can be used for assembly or adjustment.

APPLYING PRECISION ENGINEERING PRINCIPLES

It may appear that high volume production cannot reach any usable level of precision, given the constraints of manufacturing processes. However, clever design and manufacturing techniques can be used to achieve assembly precision when required. A tier-based approach to attain increasing levels of precision is presented below.

First Level

The first level of precision is accomplished by working within manufacturing processes and tolerances to produce a product with a certain level of precision. The least expensive method to obtain high levels of precision is to simply design the features into the parts themselves. The examples below would apply to molded/cast parts and formed sheetmetal.

Simple Pin and Slot

While this may seem crude by precision standards, pin and slot locating features enable very good levels of precision for low costs and at high volume production. Over-constraint is avoided and the location variation is minimized. While small cylindrical features can be molded with tight diametric tolerances, plastics can have difficulty with tolerances as the parts become large, due to material shrink during the cooling process. This can cause a pair of molded in pins to shift more than is desired. A mating hole, molded to tight diametric tolerance, provides a precise locating feature that removes two degrees of freedom in the mated assembly. Molding a mating slot, also held to tight tolerance in the width, allows parts to hold tight angular tolerances while being insensitive to material shrink. Other areas where this method is commonly used is in telecommunication

printed circuit board (PCB) backplanes, where the board needs to be precisely located for other cards to mate with it. Figure 1 shows an example of pin and slot locating features in the optical engine of a media projector.

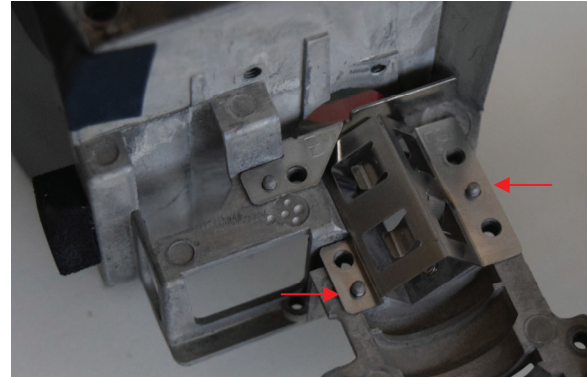


FIGURE 1. Example of pin and slot used to locate optics in a media projector (fasteners and optic element not shown).

Tapered Nesting Fits

Applying an angular contact between mating parts precisely locates them in one direction. The parts simply slide together until the gap between the tapered surfaces is closed. The nesting behavior provides exact constraint and repeatable positioning. Depending on the application and orientation, gravity can be used to keep the parts nested. This also works well for parts that need to be disassembled easily, as long as the taper angle is not low enough to for the parts to self-lock, typically when the tangent of the angle is less than the coefficient of friction between the two mating surfaces. Another factor one must consider is the distance the mating features need to separate before clearing one another (also a function of taper angle). Figures 2 and 3 show an example of a tapered locating feature used to mount a disposable drink cup dispenser to a wall.

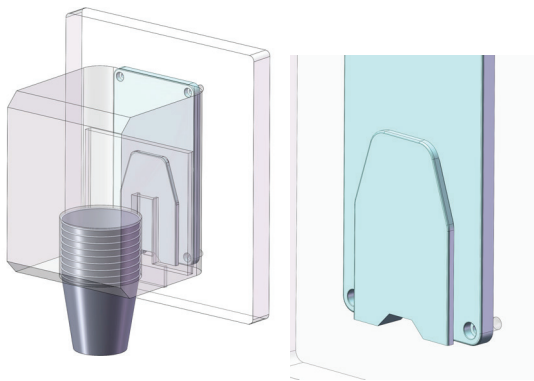


FIGURE 2. Example of tapered nesting fit in a disposable drink cup dispenser.

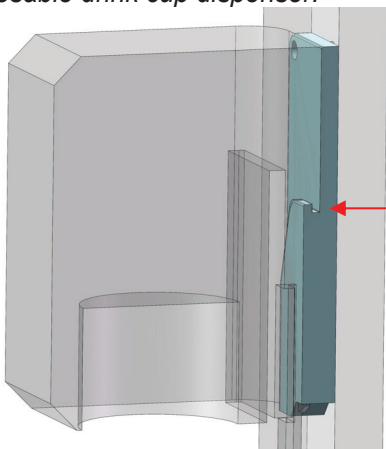


FIGURE 3. Stepped lip on wall-mounted bracket utilizing a tapered nesting fit.

Compliant locating features

When two parts need to precisely fit together a compliant locating feature on one part can enable this precision. In this case, the part is over-constrained but a compliant member is designed in and the resulting clamp force is known. An example is a molded plastic mirror holder. Molded-in features can provide a compliant force to repeatably locate the mirror without requiring additional features. Figure 4 shows an example of such a molded plastic mirror holder. One caution with molded plastic parts is their tendency to stress-relax over time, resulting in a drop in the applied force generated. Another example of a compliant locating feature is a leaf spring inexpensively stamped from sheetmetal to apply a nesting force in a product assembly.

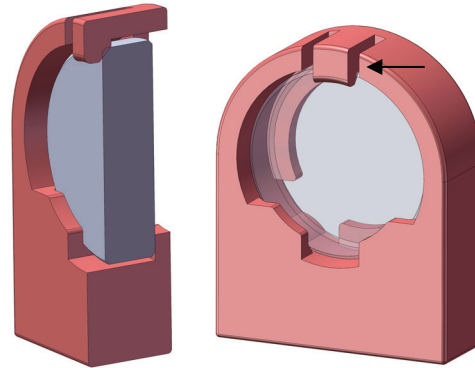


FIGURE 4. Example of a compliant locating feature in a molded plastic mirror holder ($\varnothing 12.5 \times 3.3\text{mm}$ mirror). Top “finger” applies the nesting force.

Second Level

The next level is accomplished by using assembly fixtures to improve the resulting assembly precision. Developing high tolerance assembly fixtures is a one-time capital cost that can increase the level of precision in assemblies without requiring high tolerance (and high cost) components. The fixtures hold components relative to one another while adhesives or fasteners are used to permanently secure them.

Use adhesives to constrain parts

If assembled components need to have high precision with respect to each other, an assembly fixture can be used to define the locations of the individual components. Once the components have been located and/or adjusted, a fast curing adhesive is applied to make the assembly permanent. This provides a way to avoid over-constraint since the parts are in a zero strain energy state when cured. UV (ultraviolet) curable adhesives have the benefit of long open working time with very short cure time once UV light is applied. Items to be aware of with adhesives include moisture absorption, shrink during curing, differential thermal expansion, and ultimate strength. An example of an assembly fixture used to locate an optical element before applying UV adhesive is shown in Figure 5. Figure 6 shows a close-up view of the saddle-shaped optical element and gluing features.

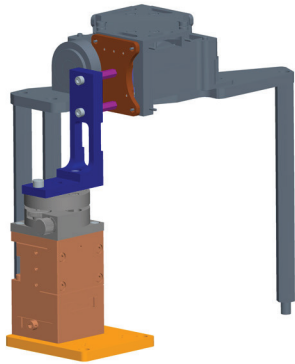


FIGURE 5. Example of an assembly fixture to precisely locate optics prior to applying UV cured adhesive.

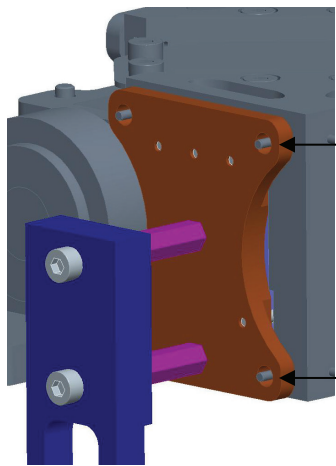


FIGURE 6. Close-up view of optical element and gluing features.

Third Level

The final level of precision is to use a secondary manufacturing process, such as machining, to obtain tighter tolerance between critical features. Sometimes a higher level of precision is required on a component than its production process can allow. This third level of precision can get around limitations such as material shrinkage during molding or casting. Many times this will require a piece of capital equipment (CNC mill) to add these features, but clever fixture design can produce similar results with low cost equipment, such as a drill press.

Machine final features into a hard disk drive housing

Disk drives are an excellent example of using multiple production methods to produce a high precision part in very high quantities. While this doesn't directly create products that avoid over constraint or deterministic assembly, it improves

the overall tolerances between features and improves surface finish, which aid in these principles. Putting multiple features into a casting enables it to serve many functions, such as locating PCBs or creating wire routing features. However, even with the best casting techniques, the precise boss diameter needed for the spinning platters and read heads requires post-cast machining. Tapped holes, locating features, and mounting surfaces can also be machined in, allowing CNC level tolerances in a simple cast part. Figure 7 shows features in a hard disk drive that were added after the housing was cast.



FIGURE 7. Machined hard disk drive housing, enabling higher levels of precision than a raw casting. Post-cast machined bosses, tapped holes, and mounting surfaces are indicated with arrows.

CONCLUSION

By using a tiered approach to apply precision engineering principles to high volume product development, the development team can control costs by using the appropriate level of precision for the product requirements.

- First Level: Start by working within manufacturing processes and tolerances to produce a product with a certain level of precision.
- Second Level: Use assembly fixtures to improve the resulting assembly precision
- Third: Use a secondary manufacturing process, such as machining, to obtain tighter tolerance between critical features

TABLE 1. High Level Comparison of a Low Volume Product versus a High Volume Product

	Low Volume Product	High Volume Product
Units per year	10s to 100s	100s to 1000s
Requirements	Performance, <u>not</u> cost	Performance <u>and</u> cost
Typical manufacturing processes	Machining, grinding, lapping (High cost)	Molding, casting, sheetmetal (Low cost)
Level of assembly adjustment	High: Use of adjustment features, shimming, lapping, external metrology	Low: Alignment fixtures, self-locating features designed into the part
Material selection	Driven by properties	Driven by properties <u>and</u> cost

Development of micro-scale bending fatigue test system for real time observation and accurate load measurement

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1. Introduction

Recently, there has been many researches and practical applications of submillimeter-submicron order structure, including the MEMS [1]. In order to improve the reliability of these structures, it is important to clarify the micro-scale material properties. Generally, material properties are considered to be constant regardless of the size of the structure. In the micro-scale, however, it is said to have different material properties in the macro-scale [2][3][4], and in particular, the fatigue behavior differs when the structure is small compared to the micro-cracks (~ 10 micron). Therefore, there are many researches to study the fatigue fracture properties of a micro structure [5][6][7], but following problems still remain.

1. Real-time observation of fatigue behavior of the specimen.
2. Accurate load measurement.
3. Simultaneous measurement of load and deformation.

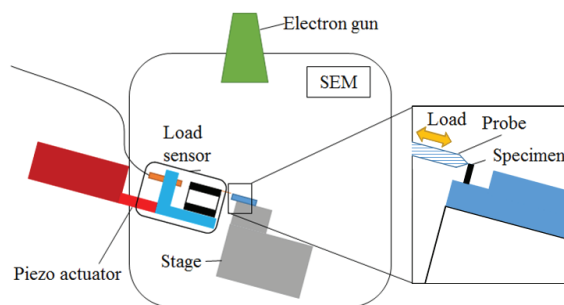


Figure 1 : Overview of the micro bending fatigue test system

To solve these problems, we proposed and developed a micro-scale bending fatigue test system that enables real-time observation and accurate load measurement (Fig.1). And with this system, we tested specimens made of single crystal silicon, which is commonly used in MEMS, etc.

2. Concepts of fatigue test system

2.1 Testing method

Generally, repeated stress that causes fatigue fracture can be classified to two types: tensile stress and shear stress. General test methods for these stresses are shown in Fig.2. Since bending loads occur the most frequently and it is impossible to observe in real-time with a rotating bending test, we chose plane bending fatigue test as main measurement.

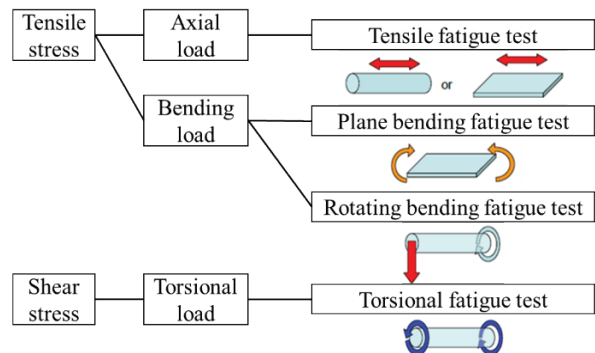


Figure 2: Fatigue testing methods

2.2 Test condition

In the fatigue test, we set the number of repetitions N to 10^6 , and the frequency of loading to 10Hz to observe specimens. The stress amplitude is assumed more than 100MPa. To observe behavior of specimen, scanning electron microscope (JSM-6301F, JEOL) was used.

3. Details of components of the system

3.1 specimen

Details of the specimen are shown in Fig.3. The specimen has a cutout portion in order to limit the fracture point. To make a wide range of stress amplitude (0.1 ~ 10GPa) possible, we prepared two types of specimens of different sizes as shown in Table.1.

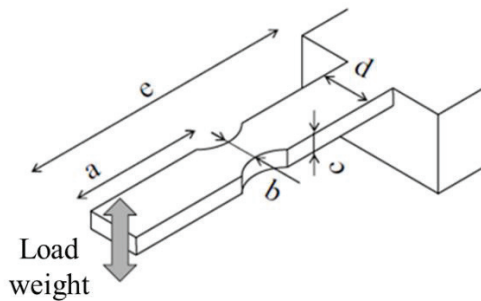


Figure 3: Details of the specimen

Table 1: Detailed size of the specimen

	a	b	c	d	e
Specimen A	15 μ m	3 μ m	2 μ m	6 μ m	25 μ m
Specimen B	15 μ m	2 μ m	1 μ m	6 μ m	25 μ m

The specimens are made of single crystal silicon, and are fabricated by Focused Ion Beam (FIB, Hitachi: FB-2000A). Fig.4 is the picture of fabricated specimen and Fig.5 shows the processes of fabricating the specimen.

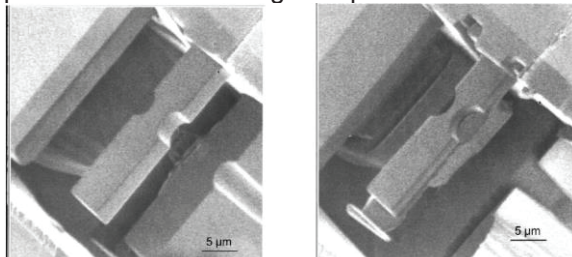


Figure 4: Images of fabricated specimens (Left: specimen A, Right: specimen B)

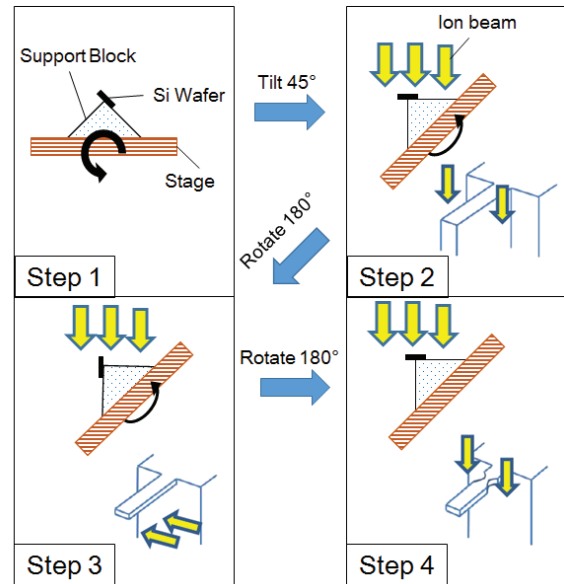


Figure 5: Fabrication processes of the specimen

3.2 Loading part

Considering the specimen size of 3.1, the resolution of the actuator is required to be approximately 100nm. Hence, in this study, a 3-axis micro manipulator with piezo actuators (Sanyu electron: SMM-7801) was used as a displacement generator. The probe is made by fabricating a stainless wire with FIB, and its diameter is less than 10 μ m at the position of 15 μ m from the tip (Figure.6). Fig.7 is the image of fabricated probe and Fig.8 shows the image of applying load to the specimen.

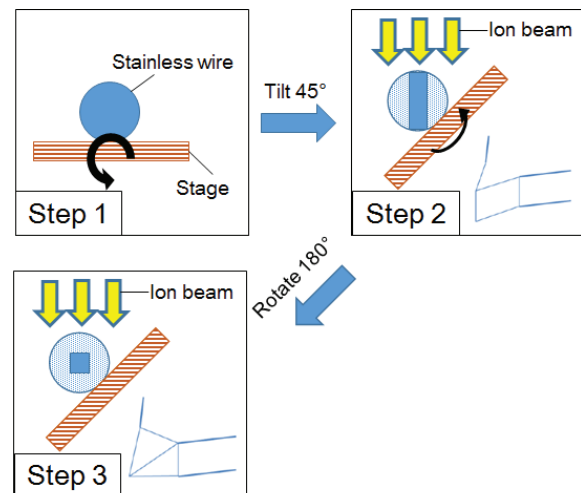


Figure 6: Fabrication processes of the probe

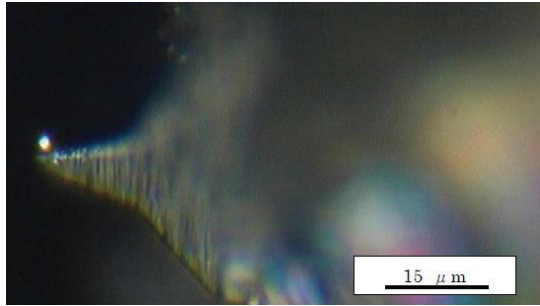


Figure7: Image of the probe

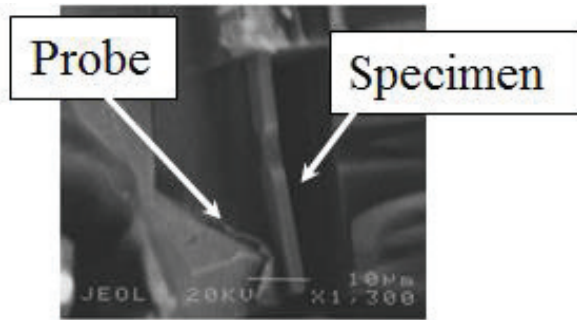


Figure8: Image of applying load to the specimen

3.3 Loading sensor

When applying the 100MPa stress amplitude to specimen A, the maximum stress is 200MPa. According to Table.1, the loading weight required to the specimen A is $26.7\mu\text{N}$. Thus, the required resolution of loading sensor should be $1\mu\text{N}$. To satisfy that, the loading sensor (right of Fig.9) consists of the parallel plates (left of Fig.9) and the eddy current sensor (Shinkawa Electric Co., Ltd.: VS -005L). Considering to the limitation of the piezo stroke, the thickness of the parallel plates was set to 0.03mm/0.06mm in order to measure larger load.

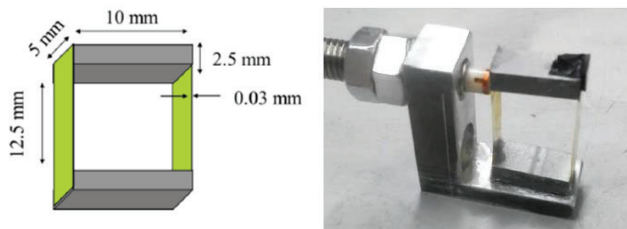


Figure 9: Detailed size and image of the sensor

According to the calibration data of these loading sensors (shown in Fig.10 and Fig.11) and the resolution of the data logger (1mV), the resolution of the sensor with 0.03mm parallel plates is $0.8\mu\text{N}$ and that of the sensor with

0.06mm parallel plates is $12.7\mu\text{N}$. Fig.12 shows the raw data of the loading sensor and displacement of the specimen A (applying the 2GPa stress amplitude in 1Hz sine wave). This figure describes that the intended loading weight is successfully applied to the specimen.

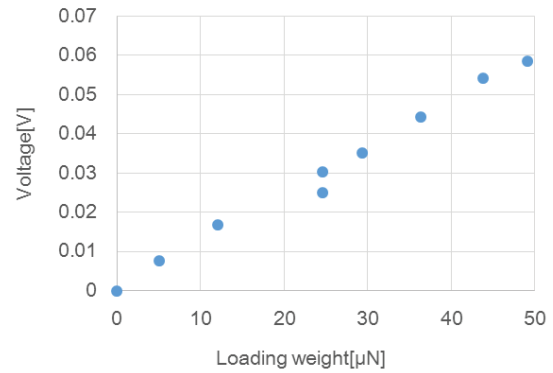


Figure 10: Calibration data of loading sensor (thickness of parallel plates is set to 0.03mm)

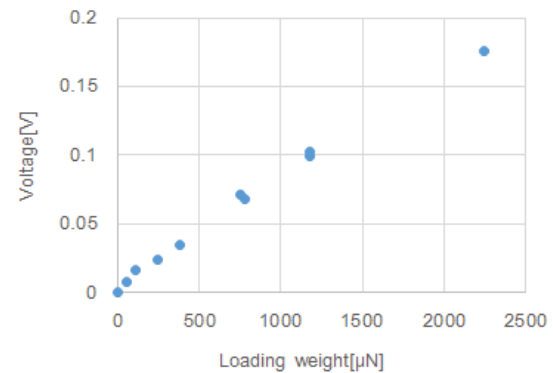


Figure 11: Calibration data of loading sensor (thickness of parallel plates is set to 0.06mm)

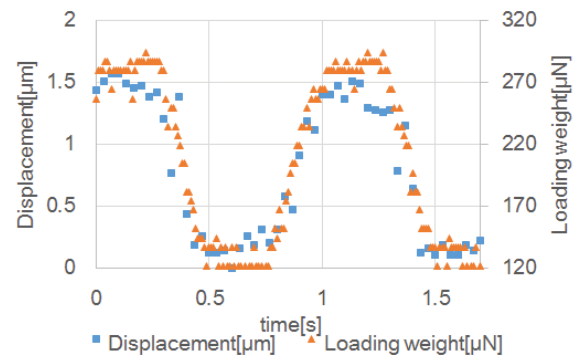


Figure 12: Raw data of loading sensor and displacement of the specimen A.

4. Bending fatigue testing of micro-scale specimen

4.1 Over view of the test

Using the developed fatigue test system explained above, fatigue tests in vacuum were conducted to investigate the fatigue properties. In addition, fatigue tests in the atmosphere were conducted as comparison experiments.

4.2 Result

Fig.13 shows the results of the test in vacuum, and Fig.14 shows the results of the tests in the atmosphere. In vacuum, the specimens were not fractured while in the atmosphere, some specimens were fractured when loading 1~2GPa stress amplitude.

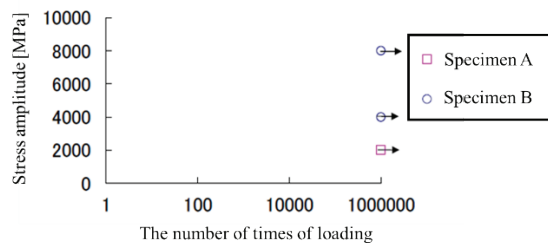


Figure13: Result of fatigue test in vacuum

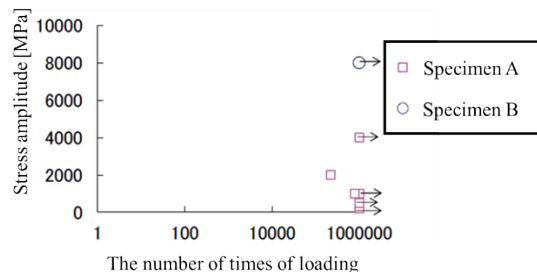


Figure14: Result of fatigue test in the atmosphere

5. Conclusion

In this study, we proposed and developed the micro-scale bending fatigue test system that meets the requirements shown in Table2.

Table 2: Required and achieve resolution

	Required resolution	Achieved resolution
Actuator	10nm	10nm
Loading sensor	1 μ N	0.8 μ N
Observation	10nm	10nm

We verified that the bending fatigue tests of micro-scale specimens can be conducted using this system with the following conclusions;

- Under the conditions in this study, the micro-scale structure made of single crystal silicon rarely causes the fatigue fracture, while in the atmosphere, it may show the fatigue fracture at relatively low stress amplitude.
- By the structural size effect, the fatigue strength of single-crystal silicon is increased, and the variation of the fatigue strength is increased at the same time.

REFERENCES

- [1] Namazu Takahiro, et al. "Fatigue Life Prediction Criterion for Micro-Nanoscale Single-Crystal Silicon Structures," Journal of Microelectromechanical Systems, vol.18, no.1, pp.129-137, 2009.
- [2] Ando Taeko, et al. "Tensile-mode fatigue testing of silicon films as structural materials for MEMS." Sensors and Actuators A: Physical 93.1, pp.70-75, 2001.
- [3] Yang, B., et al. "Stress-controlled fatigue behaviour of micro-sized polycrystalline copper wires." Materials Science and Engineering: A, 515(1), pp.71-78, 2009.
- [4] Renuart, E. D., et al. "Fatigue crack growth in micro-machined single-crystal silicon." Journal of materials research 19.09, pp.2635-2640, 2004.
- [5] Kariya, Yoshiharu, et al. "Isothermal fatigue properties of Sn-Ag-Cu alloy evaluated by micro size specimen." Materials transactions 46.11, pp.2309-2315, 2005.
- [6] Sumigawa, Takashi, et al. "Characteristic features of slip bands in submicron single-crystal gold component produced by fatigue." Acta Materialia 61.7, pp.2692-2700, 2013.
- [7] TAKASHIMA, KAZUKI, and Y. Higo. "Fatigue and fracture of a Ni-P amorphous alloy thin film on the micrometer scale." Fatigue & fracture of engineering materials & structures 28.8, pp.703-710, 2005.

THE MOTION OF A 6-DOF INCHWORM ON ROUGH SURFACE

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INTRODUCTION

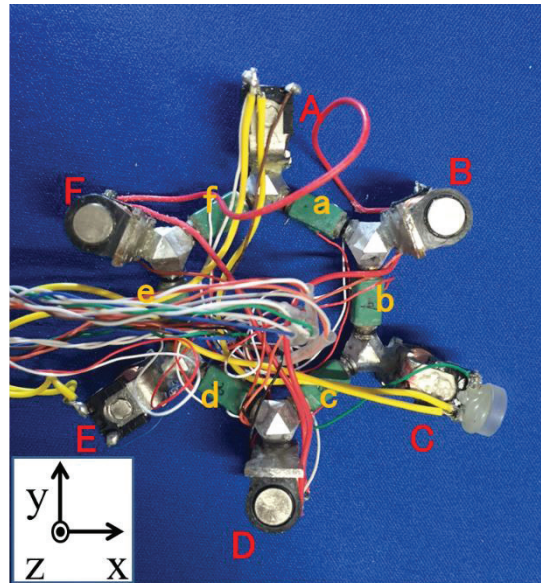
The goal of the project is the realization of a small production system using micro-actuators and these systems require sensors and actuators, which work in an unreachable small space and narrow area. Many research groups proposed and developed the actuators used in these environments. The actuators with multi-degree-of-freedom are required in the field of industrial application. We previously proposed many micro actuators consisted of piezoelectric elements and electromagnets [1]. The principle of an inchworm extends the working range of the Stewart platform [2]. The structure is similar to a cyclohexane molecule. The actuator is a conventional parallel mechanism. When we use the developed actuator in a precision positioning system, the repeatability of the actuator is one of the most important specifications in the linear and rotational displacement.

For an inchworm type actuator, a step displacement is obtained. A constant step displacement is useful for a positioning system which realizes an incremental positioning system. As the present issue, unevenness occurs to the movable displacement of the inchworm. Therefore, we try to reduce unevenness of the movable displacement by using a semispherical core into an electromagnet.

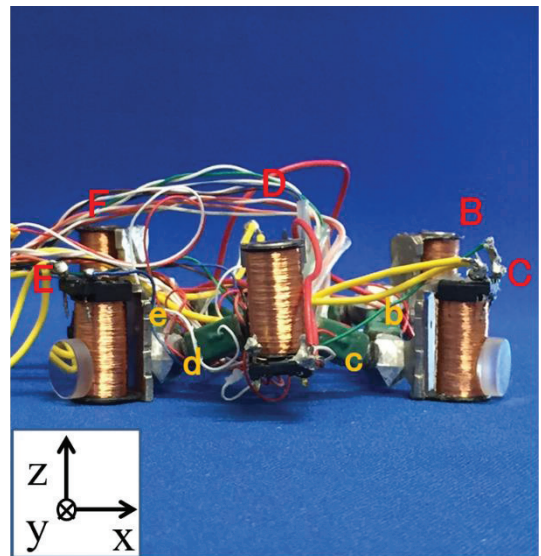
In this paper, we describe the control signals that enable the linear displacement of the actuator. We measure the displacement of two types of actuator in linear direction on four types of rough surface. One type of actuator uses flat end electromagnets and the other uses semi-spherical electro magnets. We discuss step displacement. The average for each cycle of the inchworm are discussed.

STRUCTURE OF MICRO ROBOT

Figure 1 and Figure 2 show the developed actuator. The actuator shown in Figure 1 consists of six piezoelectric elements (a-f) and six electromagnets with flat ends (A-F). The size of the actuator is 82.0 mm in width, 35.0 mm in height, and 89.0 g in weight. The actuator shown



(a) Top view



(b) Side view

FIGURE 1. Developed actuator with flat ends

in Figure 2 is similar constitution in Figure 1, but six electromagnets had semispherical ends (A-F). The size of the actuator is 82.0 mm in width, 40.0 mm in height, and 95.0 g in weight.

The actuator operates on a working surface of magnetic material. The spherical end of the electromagnets is machined in order to keep the orientation of the actuator on a surface. The piezoelectric elements deform sequentially in synchronization with the excitation of the electromagnets. The size of piezoelectric element is 6 x 6 x 10 mm. It deforms 9.1 μm in length at 150 V input. The piezoelectric elements are connected with hinge mechanisms. The feature of the actuator is that the platform and the actuator are separated. The base and the actuator are also separated. In a conventional Stewart platform, the platform, actuator, and base are linked with each other. The developed actuator realizes an inchworm motion since the electromagnet can link and unlink the actuator and the platform, and the actuator and the base.

PRINCIPLE

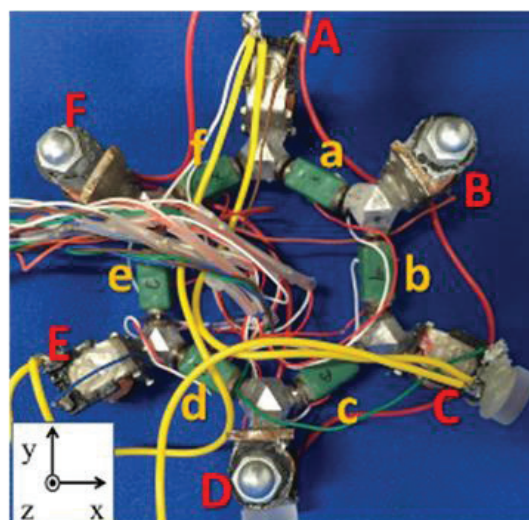
An inchworm is controlled by the operating signal. Figure 3 shows that the signal is used in movable motion. Figure 4 shows the model of linear motion. We apply the ON/OFF control voltage to the electromagnets. The voltage applied to the piezoelectric elements is determined by the inverse kinematic. The electromagnets B, D, F are excited during a control cycle, since they put the platform with the actuator.

Figure 3 and Figure 4 show the model of the linear motion mode in y-direction. In interval 1, all electromagnets are exited, and piezoelectric element b and piezoelectric element e deformed. In interval 2, electromagnet A is not excited and the electromagnet A moves in the y-direction by the deformation of piezoelectric element a and piezoelectric element f. In interval 3, electromagnet C is not excited, and the electromagnet C moves in the y-direction by the deformation of piezoelectric element b and piezoelectric element c. In interval 4, electromagnet E moves. In interval 5, the actuator returns to the initial condition by the piezoelectric elements returning to the initial length. The actuator can move in linear direction.

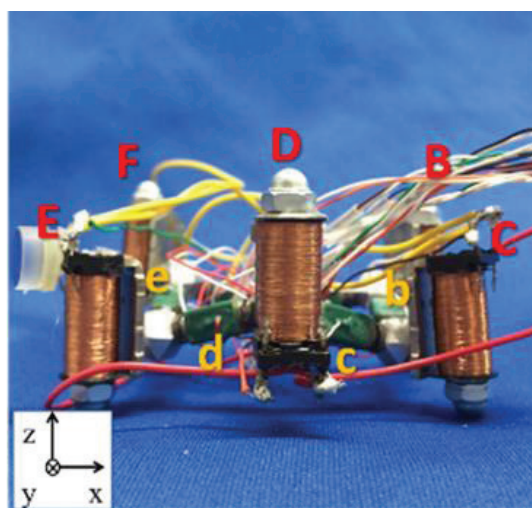
EXPERIMENT

Figure 5 shows the experimental setup used in the measurement of the linear motion.

The inchworm is placed on four types of rough



(a) Top view



(b) Side view

FIGURE 2. Developed actuator with semispherical ends

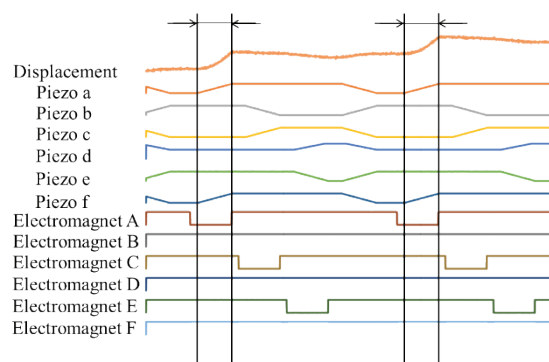


FIGURE 3. Signal of movable linear motion

surface. The control signals shown in Figure 3 are generated by a personal computer and applied to piezoelectric elements and electromagnets through an amplifier. The inchworm realizes a linear motion by the use of control signals. The linear displacement are measured by displacement meters which measure the displacement in the y-direction of the measurement points A, C and E. Measurement targets are small optical mirrors fixed on electromagnet A, C and E. The linear displacement measurements are repeated five times. The maximum voltage applied to the piezoelectric element is 150 V, and that applied to the electromagnet is 10 V. The deformation of the piezoelectric actuator is $9.1 \mu\text{m}$ at 150 V. The electromagnetic force of the semi-spherical end electromagnet is 0.06 N and the electromagnetic force of the flat end electromagnet is 1.18 N at 10 V. The control cycle is 1 s which includes all intervals shown in Figure 3 and the control signal is applied for five cycles.

EXPERIMENTAL RESULTS

Figure 6 shows y-direction displacement for five cycles, when the actuator operated in the movable linear motion. The inchworm using semi-spherical end electromagnets A, C and E direction on rough surface $1.0 \mu\text{mRa}$. We measured the displacement five times. The average displacement for 5 s is shown in Figure 7. The displacements of the measurement point A, C and E are plotted. Although the displacement per cycle differs from one another, all results show the similar trend. The experimental results are difference between Figure 7 (a) and Figure 7 (b). Figure 7 (a) shows the linear displacement of the inchworm using flat end electromagnets and semi-spherical end electromagnets. The positions of three electromagnets A, C, E are plotted. These electromagnets are on the operation surface. The surface roughness used in the experiment is $1.0 \mu\text{mRa}$. When the flat ends electromagnets are utilized, the velocity was about $6.60 \mu\text{m/s}$ of A, about $2.79 \mu\text{m/s}$ of C and was about $5.27 \mu\text{m/s}$ of E shown in Figure 7 (a). The use of the semispherical end electromagnets however shows the similar velocity, almost $5.0 \mu\text{m/s}$ in Figure 7 (b).

We repeated the displacement measurement for 5s using different surface roughness of 0.5, 1.5 and $2.0 \mu\text{mRa}$, and the results are summarized in Table 1, which corresponds to the velocity in Figure 7.

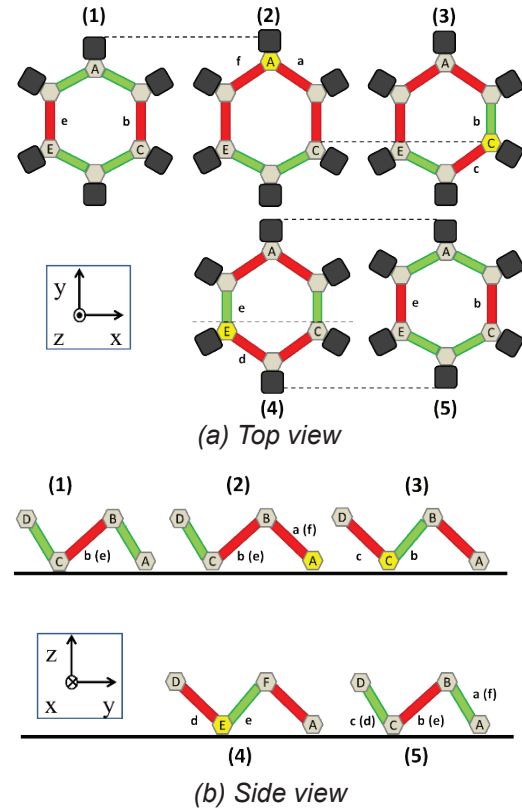


FIGURE 4. Model of the movable linear motion

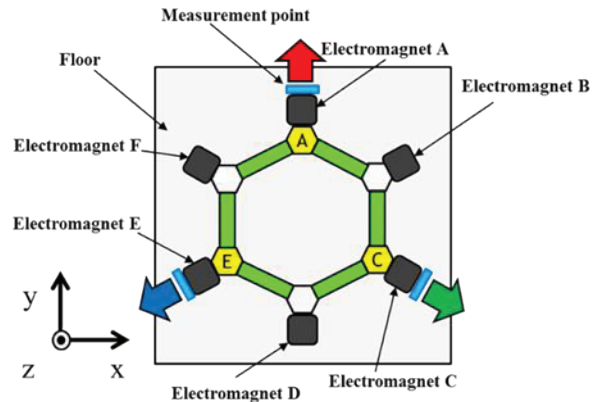


FIGURE 5. Experimental environment

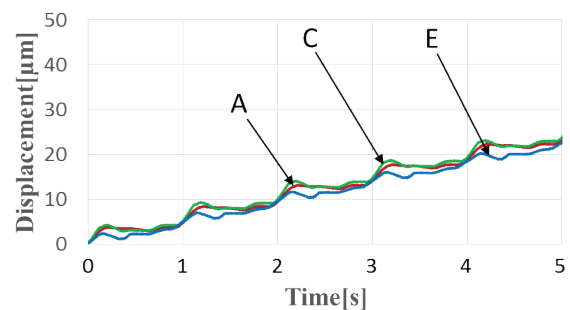
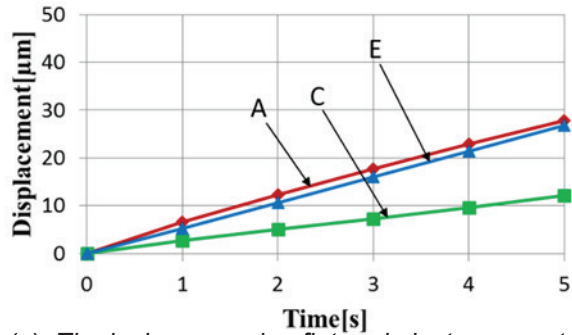
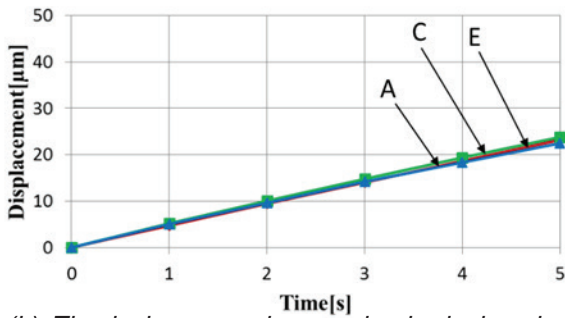


FIGURE 6. y-direction displacement for five cycles

The displacement of electromagnet A, C and E can approximate to each other by the electromagnet with semi-spherical end core. These results indicate that the actuator proceeds to the y-direction with the interval of 1 s. The difference between the trials may be caused by the misalignment of the measurement sensors.



(a) The inchworm using flat end electromagnets



(b) The inchworm using semi-spherical end electromagnets

FIGURE 7. y-direction displacement for five cycles

SUMMARY

In this paper, we described the displacement of two type of inchworm on rough surface. We confirmed the actuator was able to move by the principle of an inchworm. The linear displacement of the inchworm using flat end electromagnets on $1.0 \mu\text{mRa}$ for the directions A, C, E were about $27.8 \mu\text{m}$, $12.1 \mu\text{m}$, $26.8 \mu\text{m}$. However, the inchworm using semi-spherical end electromagnets on $1.0 \mu\text{mRa}$ for the directions A, C, E were almost $23.0 \mu\text{m}$. We found differences in the amount of displacement of each operation. Each of y-direction displacement for five cycles was different when the actuator was operated in the movable linear motion. The object position measuring apparatus is considered when we use the actuator in precision positioning system. In every trial, however, equal step displacements are obtained with several exceptions. From the average step of the displacement, the actuator is used in the precision positioning in near future. This research project was financially supported by the Japan Society for Promotion of Science (JSPS), Grants-in-Aid for scientific Research (C), No. 26420212

REFERENCES

- [1] R. Kamiya, A. Torii, K. Doki, A. Ueda "Performance of a Movable Stewart Platform Using Piezoelectric Actuator", Proceedings of the 5th International Conference on Positioning Technology, pp.25-26, 2012.
- [2] T. Yamada, A. Torii, A. Ueda "Positioning of an inchworm Type Actuator with Five Degree of freedom", Int. Symposium on Micro/nano mechatronics and Human Science (MHS) 2011 pp.267-272

TABLE 1. Displacement of electromagnet A, C, E for 5 s

Surface roughness (μm)	Electromagnet	Displacement with flat core (μm)	Displacement with semispherical core (μm)
0.5	A	7.7	15.6
	C	24.4	17.0
	E	38.2	18.1
1.0	A	27.8	23.2
	C	12.1	23.7
	E	26.8	22.5
1.5	A	15.9	0.2
	C	4.7	0.5
	E	15.5	0.8
2.0	A	28.4	0.4
	C	1.4	0.6
	E	24.0	0.3

TOOL POSITION COMPENSATION FOR MACHINING A MICRO-PRISM PATTERN ON A SINUSOIDAL SURFACE BY SHAPING

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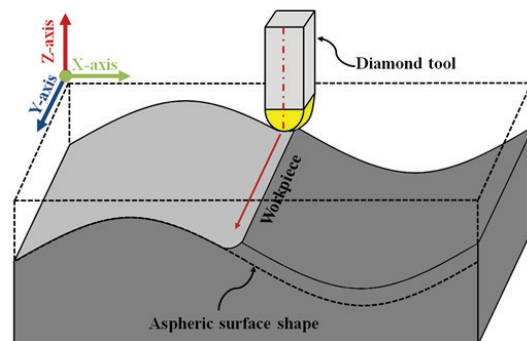
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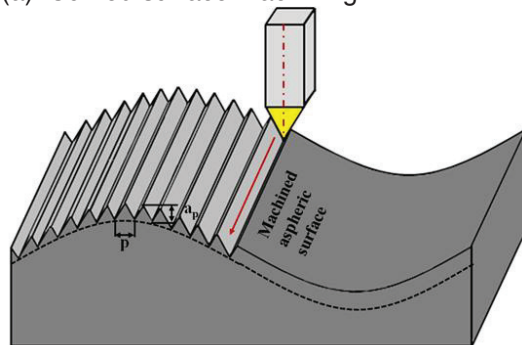
INSTRUCTIONS

A light guide plate (LGP), which is a component of display devices for luminance and uniform illumination, is composed of prism sheets and diffusion sheets. The LGP has been developed to increase the light efficiency while decreasing the amount of optical sheets to provide slimness, reduced weight, and cost reduction. Numerous studies on LGPs with micro-patterns have been performed to develop functions. Recently, in addition to LGPs having flat surfaces, light guide plates with curved surfaces also have been studied. Molds for LGPs are generally machined by using a mechanical method. With this approach it is possible to easily machine metal molds with appropriate surface roughness. Figure 1 shows a micro-pattern mold machining method on a curved surface by an ultra-precision shaping machine. The curved surface is machined by a round tool, and then the tool is changed to a pattern tool. A micro-pattern is then machined on the curved surface. However, two problems should be considered when molds with micro-patterns on curved surfaces are machined using an ultra-precise shaping machine tool. First, interference between the round tool and the curved surface occurs. A tool path is generated based on the center of the edge of the round tool when machining the curved surface. Interference occurs due to the shape of the round tool. Thus, a tool path should be generated with no interference. Second, if the tool is changed to machine the micro-pattern on the machined surface, a position error between the changed tool and curved surface occurs. The position of the edge of the pattern tool is always changed because the tool is manually installed by the user. The changed tool thus should be synchronized to the curved surface machined by the round tool. These two processes should be performed in order to machine the micro-patterns precisely on the

curved surface. In the present study, a sinusoidal surface was machined with minimized interference, and a micro-pattern was machined by using a changed tool that was synchronized to the curved surface.



(a) Curved surface machining



(b) Prism pattern machining

FIGURE 1. Prism pattern machining method on the curved surface

SINUSOIDAL CURVED SURFACE MACHINING METHOD AND PRISM TOOL SYNCHRONIZATION METHOD

First, sinusoidal machining technology is necessary to machine the prism pattern on the sinusoidal curved surface. The main problem with sinusoidal surface machining is interference with the round tool and the curved surface.

The interference with the tool and the designed curve were geometrically analyzed, as shown in Figure 2. The machined curve has an error with the designed curve due to the radius of the tool. Analysis of this error revealed that error of d occurs. d_1 and d_2 , which are functions of θ and the tool radius (r), should be calculated in order to calculate d . Here, θ is the gradient of the curve, and it can be calculated by the differential of curve function. When d is calculated by d_1 and d_2 , the error can be compensated.

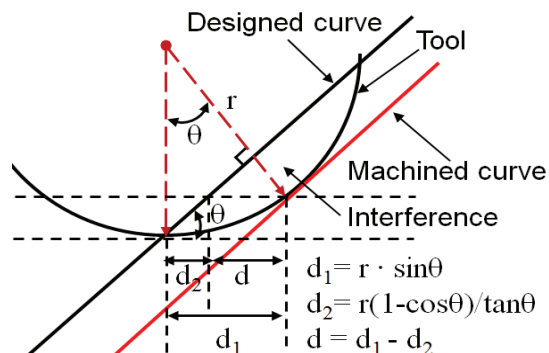


FIGURE 2. Geometrical analysis of the interference between the round tool and the curved surface

The machined sinusoidal surface and the changed tool were synchronized by using a tool dynamometer. The point that could be easily contacted by the pattern tool was the minimum point of the sinusoidal curve, as shown in Figure 3. First, the tip of the tool was located in the concave part of the sinusoidal surface in order to find the minimum point (I). Both contact points were found after the tool was moved from side to side along the X direction (II). The X position of the tool was matched to the minimum point as the tool moved to an intermediate/the middle point of both contact points (III). The tool was matched to the minimum point as the tool moved along the Z direction. The machined sinusoidal surface and the changed tool were synchronized by the minimum point (IV).

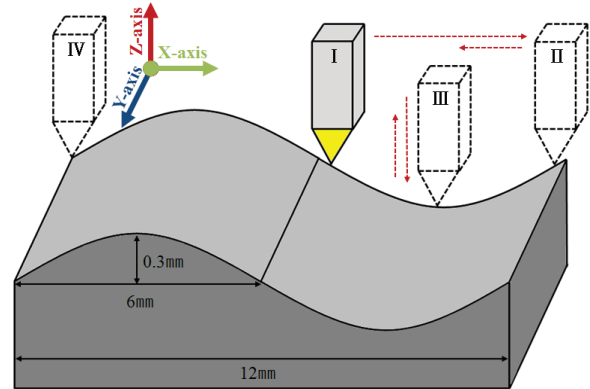


FIGURE 3. Synchronizing method between changed tool and sinusoidal curved surface

SINUSOIDAL PATTERNS MOLD MACHINING SYSTEM

Prism pattern machining on the sinusoidal curved surface was performed on an ultra-precision machine tool system, as shown in Figure 4. This system was composed of X, Y, and Z axes, and had a position resolution of 10 nm. A tool dynamometer that could measure cutting force as low as 0.002 N was installed on the Z axis; the dynamometer thus could be used to measure tool contact. The workpiece was fixed by a vacuum jig. The radius of the round tool is 5 mm, and a 90° diamond tool was used, as shown in Figure 5.

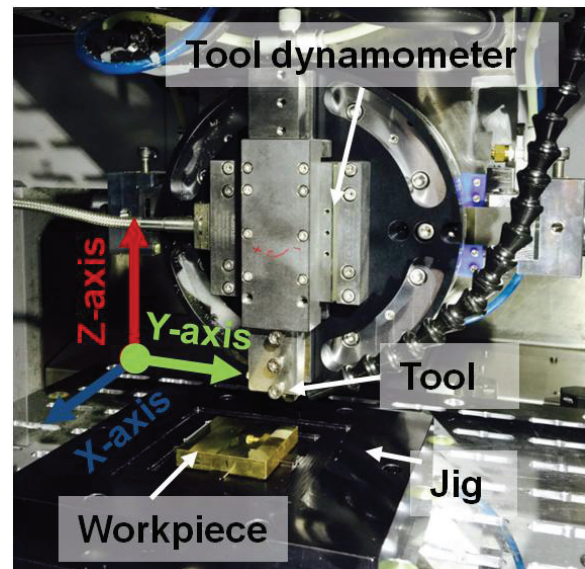


FIGURE 4. Prism pattern machining system on the sinusoidal surface

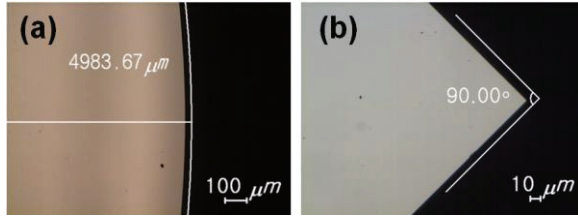


FIGURE 5. Tool for machining prism pattern on the sinusoidal surface (a) R 5mm (b) V90°

SINUSOIDAL SURFACE MACHINING RESULT WITH COMPENSATION

A sinusoidal shape was machined by using this method. The sinusoidal shape had a period of 6 mm and amplitude of 0.6 mm. The tool radius is the most important factor because it directly affects the shape error. Although a tool radius of 10 mm is generally used to machine a flat mirror surface, a diamond tool with a radius of 5 mm was employed in this experiment. The specific cutting conditions are given in Table 1.

TABLE 1. Sinusoidal surface cutting conditions

Items	Conditions
Cutting tool	R5mm diamond tool
Workpiece	64 brass 50 x 50 mm ²
Pitch	Roughing 50 μm and finishing 30 μm
Depth of cut	Roughing 20 μm x 30 and finishing 10 and 3 μm
Feedrate	12,000 mm/min
Cutting oil	Kerosene mist

Figure 6 shows the machined mold with a sinusoidal surface. The sinusoidal surface appears to be clearly machined. The machined mold with a sinusoidal surface was measured by a surface profiler (Form Talysurf PGI 1240) as shown in Figure 7. The period error most highly occurred at the inflection point. The error without compensation was 0.394 mm, and the error with compensation was 0.005 mm. Through the present study, the compensation method was thus proven to be effective.

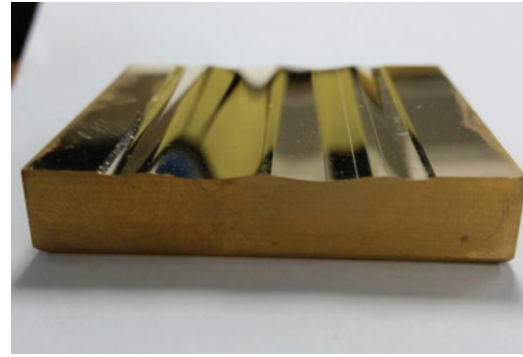


FIGURE 6. Machined mold having sinusoidal patterns with a compensated tool path

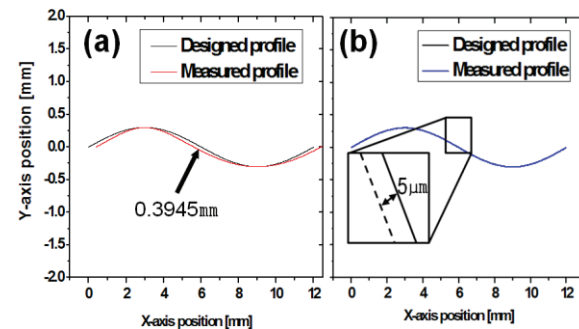


FIGURE 7. Profile of machined mold having sinusoidal surface (a) without compensation and (b) with compensation

PRISM PATTERN MACHINING ON THE SINUSOIDAL SURFACE

A prism pattern with a pitch of 40 μm was machined on the sinusoidal surface with a period of 12 mm and amplitude of 0.6 mm. The prism tool and the sinusoidal surface were synchronized using the synchronizing method suggested in the present study. Overdepth was applied to minimize the contact error, although the depth of cut was 20 μm. The specific cutting conditions of the prism pattern are given in Table 2.

TABLE 2. Specific prism pattern cutting conditions

Items	Conditions
Cutting tool	V 90° diamond tool
Workpiece	Machined workpiece
Pitch	40 μm
Depth of cut	10, 7, 3, 2 and 1 μm
Feedrate	12,000 mm/min
Cutting oil	Kerosene mist

The machined mold with a prism pattern on the sinusoidal surface was analyzed by SEM, as shown in Figure 8. Although slight burrs were

observed, the prism pattern was entirely machined on the sinusoidal surface. The size of the pitch of the prism pattern was $39.9\ \mu\text{m}$, as determined using a microscope and as shown in Figure 9. The prism pattern was precisely machined on the sinusoidal surface. The proposed tool synchronizing method was thus found to be effective.

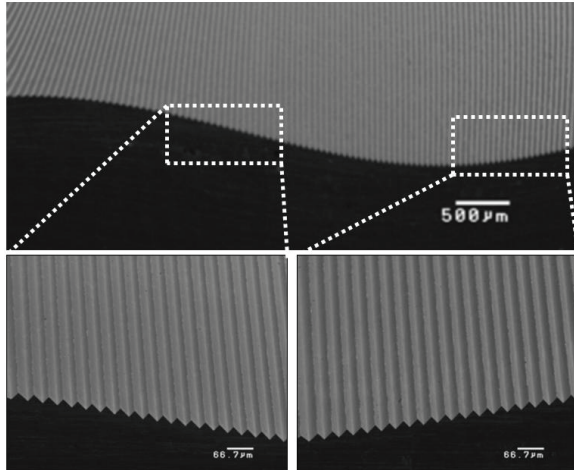


FIGURE 8. SEM image of machined micro-prism pattern on sinusoidal surface

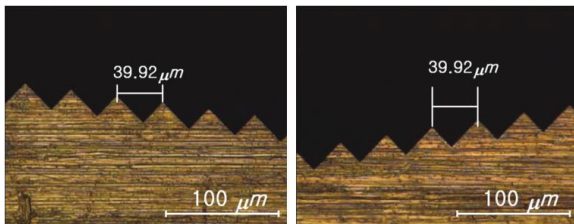


FIGURE 9. Size of machined micro prism pattern on sinusoidal surface

CONCLUSION

In the present study, the problems that arise when machining a prism-pattern on a sinusoidal surface were analyzed. The interference with the tool and the curved surface when machining the sinusoidal surface was geometrically analyzed and minimized. A method to synchronize the changed tool to the curved surface using a tool dynamometer was suggested. Finally, a prism pattern with a pitch of $40\ \mu\text{m}$ was precisely machined on the sinusoidal surface with a period of 12 mm and amplitude of 0.6 mm.

REFERENCES

- [1] Lindeke R, Schoenig F Jr, Khan A, Haddad Joo-Hyung Lee et al. Design and fabrication of a micropatterned polydimethylsiloxane (PDMS) light-guide

plate for sheet-less LCD backlight unit. *Journal of the Society for Information Display*, 2008;16(2): 329-335.

- [2] T. Nishiguchi et al. Improvement of productivity in aspherical precision machining with in-situ metrology. *CIRP Annals - Manufacturing Technology*, 1991; 40(1): 367-370.
- [3] Yung-Tien Liu et al. A study on optimal compensation cutting for an aspheric surface using the Taguchi method. *CIRP Journal of Manufacturing Science and Technology*. 2010;3(1):40-48.
- [4] Hyun-Chul Kim et al. Development of machining technology for micropatterns with large surface area. *International Journal of Advanced Manufacturing Technology*, 2012;58(9-12):1261-1270.
- [5] Eun-Suk Park. A study on micro pattern machining on aspheric surface using orthogonal planning method. Master degree. University of Science & Technology, 2013.
- [6] F. J. Chen et al. Profile error compensation in ultra-precision grinding of aspheric surfaces with on-machine measurement. *International Journal of Machine Tools and Manufacture*, 2010;50(5): 280-486.
- [7] Hwan-Jin Choi. Improvement of machining precision by using compensation of tool position of micro variable angle prism patterns. Master degree. University of Science & Technology, 2013.
- [8] Tae-Jin Je et al. Ultra-precision sinusoidal patterns machining using a planning machine. *Proceedings of American Society for Precision Engineering*, 2014: 521-524
- [9] Tae-Jin Je et al. Compensation method of machine shape error according to tool radius and curvature of curved surface in shaping. *Proceedings of International Conference of Manufacturing Technology Engineers*, 2014: 26

Comparison of various trigger touch probes

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INTRODUCTION

Touch probes are an essential part in the calibration of accuracy and the use of high precision machining centers. The requirements of 5-axis machining centers are constantly increasing. Simultaneously the accuracy demands of touch probes are rising. Therefore touch probes have a variety of several tasks.

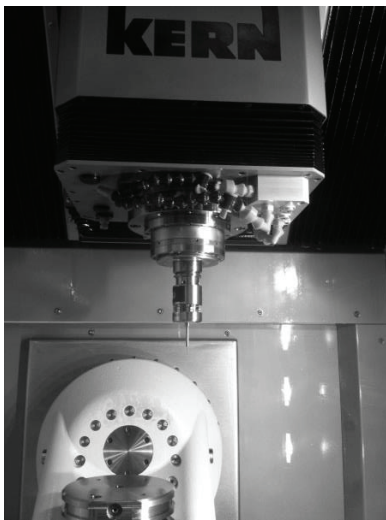


FIGURE 1. Touch probe on "Kern Micro"

Various tasks of touch probes

Dimension and position of raw material

The reference point of the workpiece gets determined. Furthermore the precise work piece dimensions are determined [1].

Checking accuracy of the processing results

The touch probe is used as an inspection to determine if the machined dimensions have been complied. The rising machining accuracy needs to be considered. There is a relationship between machining accuracy and accuracy of measuring equipment. Due to this relationship, the accuracy requirement of touch probes is going to increase as well.

Identification of machine kinematics

This type of application represents the highest requirement. The demands going to increase steadily due to rising accuracy expectations of

5-axis machining centers. The 5-axis machining centers have swivel axes. In order to receive a workpiece with high accuracy the machining center has to know the exact position of the swivel and rotation axes [2].

A possibility to determine the position of the axes is given by touch probes. The error of the touch probes affects the position detection of the swivel axes. For this purpose illustrative example is followed. The Figure 2 shows that the position of a swivel axis has a significant error influence.

In the figure the rectangular workpiece can be seen. At point A you can see the identified swivel axis. So the actual physical swivel axis is located at point B. Due to this, there is an error in the determination of the swiveling center (e_x). If the workpiece is rotated by 180° , you will get a doubling error ($2 e_x$). This happens because the machine expects the workpiece at point C. However the workpiece is located at point D. If the workpiece is rotated less than 180° , the error gets added vectorial.

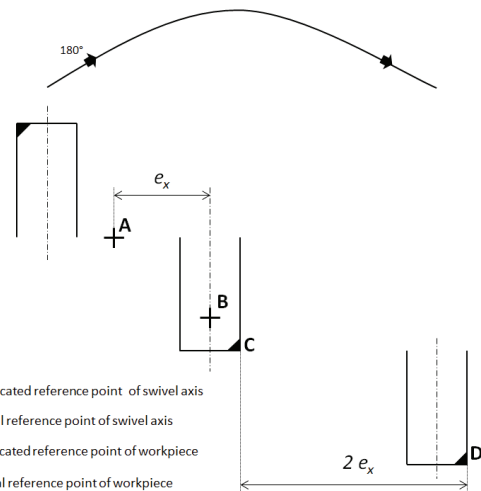


FIGURE 2. Rotating workpiece by moving swivel axis of 5-axis machining center

You can say that the identification of machine kinematics depends on the accuracy of the touch probe. So the identification of machine kinematics of 5-axis machining centers is limited in terms of the accuracy of touch probes.

REQUIREMENTS OF TOUCH PROBES

Reproducibility

The reproducibility of a touch probe should be smaller than $\pm 0.25 \mu\text{m}$ ($\pm 19.7 \mu\text{in}$). It remains to determine the target fault to be able to produce more accurate workpieces. The reason for this is that the reproducibility of the touch probe must be lower than the position variation range of the linear axes. The “Kern Pyramid Nano” has a position variation range of $0.3 \mu\text{m}$ at a deviation of $\pm 3\sigma$.

Trigger forces of touch probes

Generally, there exist three problems with contact forces. First, the trigger force should not be too small due to the sensitivity against oil, - and coolant films. The second problem is that the forces also shouldn't be too large. This can lead to elastic displacements and plastic deformations. Therefore it is important to find the right compromise of the trigger force. Finally, the trigger forces have noteworthy differences in individual directions. The current state of technology is that the displacement in the z-direction is approximately 10 times greater than in x, y- direction. Thereby pre-travel-variations going to occur in the case of 3D-level measurements. Due to this, the calibration is much more complicated.

EXPERIMENTS

To examine how the trigger touch probes accomplish the requirements two types of experiments have been carried out. During the tests six different types of touch probes have been evaluated on two different machining centers.

Contrary to other examinations [3] [4] the aim of this analysis is not to split the characteristics from the probes to the properties of the machine. Rather to get a result of the performance on the machine. All in all we get a result of the machines with their probes.

Further the measurements are executed with following setup for the trigger probes (FIGURE 3, TABLE 1):

TABLE 1. Measurement setup

length stylus	40 mm
diameter stylus tip	1 mm
probing speed v	200 mm/min

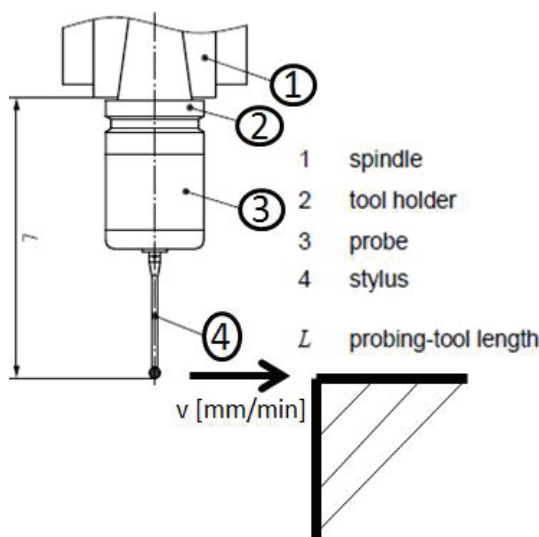


FIGURE 3. Measurement setup [3]

Characterization of trigger mechanism

Different probe suppliers have different mechanical designs of the probes' measuring mechanism [5]. Different designs mean different pre-travel. Through this experiment, the pre-travel variation (PTV) could be determined in the x, y plane. The measured value depends on the trigger direction. For the measurement, a 5-axis machining center (Kern Micro) is used. According to ISO 230-10 [3] the characterization is measured by using a calibrated ring as target. The roundness of the target is better than $1 \mu\text{m}$. Starting with 0° , which means probing the positive x-direction (FIGURE 4). The inside of the ring is sampled in 5° increments contra clockwise (72 points in the inner shape). Each point is measured 5 times. The touch probe is adjusted to concentricity smaller than $2 \mu\text{m}$ and calibrated in x, y-direction.

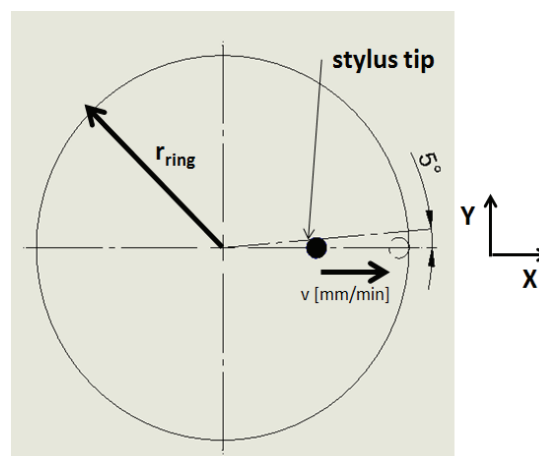


FIGURE 4. Ring target for characterization of trigger mechanism

Result:

To compare the different PTV of the different types of probes the measurements are shown in a circular diagram (FIGURE 5.)

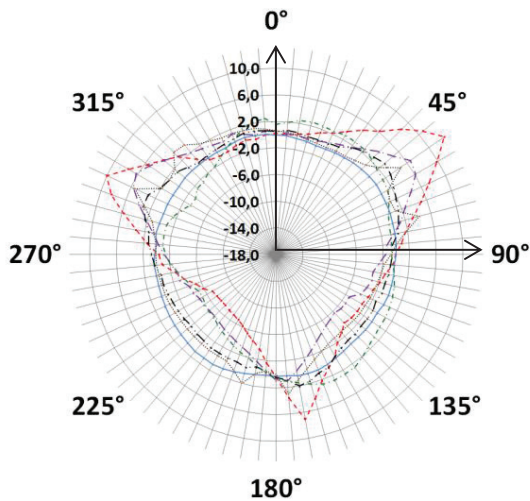


FIGURE 5. Polar plot characterization of PTV of several touch probes

In FIGURE 5 you can see the average deviance of five repetitions from the measured radius of the ring to the nominal radius by the different touch probes. The angle in FIGURE 5 is the direction of the measurement and the radial lines the mean of the PTV [μm]. For an assessment the maximum of the pre travel variation and the standard deviation over the whole target is used.

	σ_R [μm]	Max. PTV* [μm]
--- Probe 1	4,81	20,0
— Probe 2	0,38	1,6
— Probe 3	3,36	12,2
— Probe 4	1,3	4,8
--- Probe 5	2,4	8,0
.... Probe 6	1,84	8,3

PTV: Pre-Travel-Variation
 σ_R : standard deviation

FIGURE 6. Result of PTV and standard deviation

As you can see in FIGURE 6 there is no probe without a pre-travel smaller than $1 \mu\text{m}$. Further it is not recommended to use the probes to measure sizes smaller $1 \mu\text{m}$ whit this setup. A way to handle this is to orientate the probe. Orientation means that the touch probe gets rotated simultaneously to the polar coordinates

of the target (FIGURE 7). Consequently the probes measuring mechanism gets strained in the same direction for each angle of the ring. This modification is used as standard process on KERN machines.

2D/3D determination of reproducibility

The following measurements are executed with orientated probes, to get the focus on repeatability. Moreover the examination is done according to ISO 230-10 [3].

2D-reproducibility

The second type of measuring is to determine the reproducibility in the 2D level. The process almost has the same procedure like the determination of the trigger characterization. Differences are the machine, the number of samples and the gap size of the measurement points. To get even more accuracy in the positioning a hydrostatic 5 axis machining center (Kern Pyramid Nano) is used. The target is measured in increments of 10° (FIGURE 7). Every point out of 36 points is sampled 25 times. More measurements were needed to give a proper statistical statement about the reproducibility.

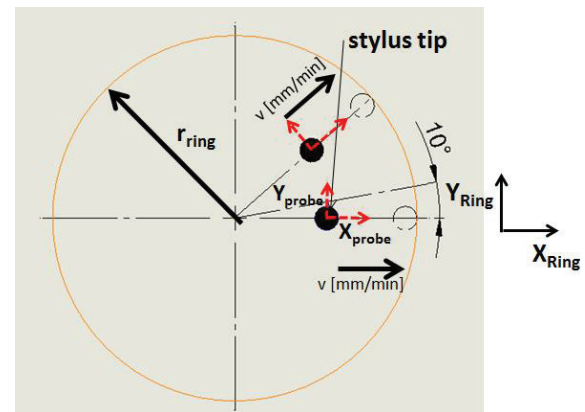


FIGURE 7. setup for 2D-reproducibility

Result:

To compare the reproducibility a bar chart is used (FIGURE 8)

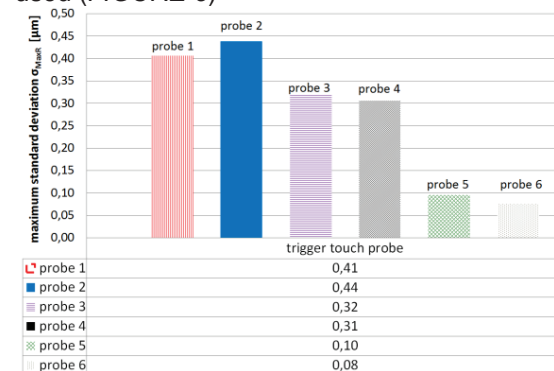


FIGURE 8. 2D-reproducibility of evaluated touch probes

At the bar chart on *FIGURE 8* you can see the maximum standard deviation out of 25 repetitions on the y-axis. Maximum standard deviation means, the worse cased probed angel gets assessed. Different probes show different sizes of reproducibility. This is caused by different mechanical designs of the trigger mechanism. According to the requirements of trigger touch probes the high reproducibility is very important for machine calibration.

3D-reproducibility

Trigger touch probes are usually calibrated in the xy-plane and in z-direction. So on we know the effective stylus diameter in these directions. Further the focus of their mechanical design is on measuring in a plane or in z-direction. To get an assessment about the behavior of trigger touch probes for 3D-measurements this experiment is carried out. The focus of this experiment is to earn knowledge about the reproducibility for measurements in 3D areas. For the measuring a polished sphere with nominal diameter 20 mm (grade 5) is used as setup. The center coordinates of the sphere are measured by probing five points for each probe. Four points in xy-plane and one point in z-direction. To get a reliable center of the sphere the average of five repetitions is used. To detect the center in z-direction the probe is calibrated in this direction. For the measurement the sphere is described by longitudes and latitudes [4]. Starting at the equator the sphere is probed in 10° steps in latitude direction and in 15° steps in longitude direction (*FIGURE 9*). Each coordinate is probed in vector direction with 25 repetitions.

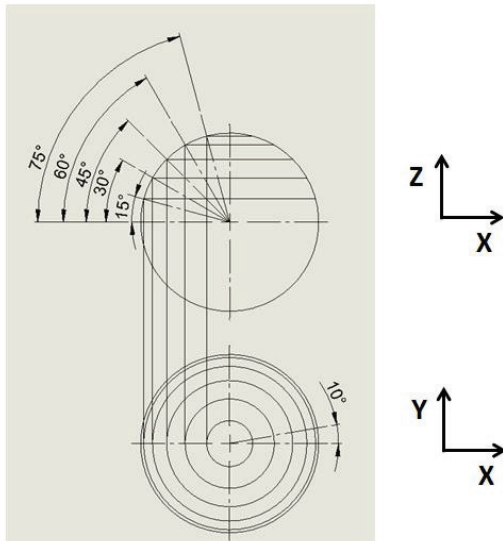


FIGURE 9. Target for 3D-reproducibility test

Result:

For each probing point the radial distance from the sphere center is calculated with the square

root of the sum of the squares of the measured X-, Y- and Z-axis coordinates. For an assessment in not used the deviation from the nominal radius to the measured radius, it's used the reproducibility out of 25 repetitions. In *FIGURE 10* a comparison of the standard deviation of the different touch probes is show.

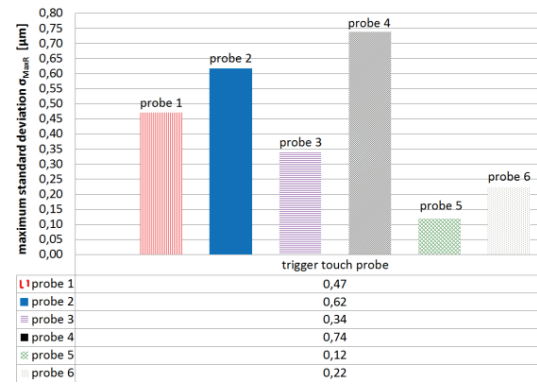


FIGURE 10. 3D-reproducibility of evaluated touch probes

As for the 2D-reproducibility a bar chart (*FIGURE 10*) is used to picture the maximum standard deviation. Maximum standard deviation for the 3D-reproducibility means to assess the sphere coordinate with the highest score of deviation. In applications this is an important value for probing freeform surfaces. The different trigger touch probes show different results. Similar to the 2D-reproducibility measurements these differences depend on different types of the mechanical designs of the trigger mechanism.

CONCLUSION

The different touch probe trigger systems show a huge difference in the pre-travel variation, depending on the design of the trigger mechanism. Also at the best systems an orientation of the system in probing direction is still reasonable.

In 2D-measurements a reproducibility of the hole system (machine and trigger touch probe) below 0.1 μm is possible. In 3D-measurements 0.1 μm is also possible. There is only one system, with can reach this reproducibility. For the state of the art of machine tool calibration this values are useable.

QUESTIONS

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REFERENCES

- [1] Renishaw GmbH, „Der Bessere überlebt - Prozessüberwachung in der spanenden Bearbeitung,“ Renishaw, Pliezhausen, 2011.
- [2] HEIDENHAIN, "Technisches Handbuch iTNC530," HEIDENHAIN, Traunreut, 2010.
- [3] ISO, Determination of the measuring performance of probing systems of numerically controlled machine tools, Switzerland: ISO, 2011.
- [4] A. W. M. B. Michal Jankowski, "Machine toll probes testing using moving immer hemispherical mater artefact," ELSEVIER, Warsaw,Poland, 2014.
- [5] Renishaw GmbH, „Rengage_3D_Messtechnik 03,“ Renishaw, Pielzenhausen.

MICRO-LASER ASSISTED DRILLING OF SINGLE CRYSTAL SILICON IN DUCTILE REGIME

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INTRODUCTION

Micro drilling is an established technology, but with many limitations. The techniques and processes vary widely depending upon the material being drilled, precision and accuracy required, size (diameter, depth, etc.). Current applications demand better quality materials that are mechanically harder, yet manufactured with high level precision and accuracy. Drilling free of fractures, surface and subsurface damage and cracks and micro-cracks with good edges and high surface quality of the brittle and hard materials such as ceramics and semiconductors is always a challenge due to their low fracture toughness. Often, severe fracture can result due to their low fracture toughness. For silicon, as an example, fracture toughness is 0.83 to 0.95 MPa.m^{0.5} (depends on crystal orientation).

There are many micro drilling techniques available to generate micro-holes such as micro-electrical discharge machining (micro-EDM), laser, photo-etching, ultrasonic and micro-electrical chemical machining (micro-ECM). However, each process has its limitations in cost, machining efficiency, properties of the workpiece and aspect ratio of a micro-hole [1,2]. In this current work a method of drilling by using laser which is focused through a diamond to the tip of the tool to drill hard and brittle materials is presented.

In the previous research works [3-6], it has been demonstrated that the ductile mode machining of semiconductors and ceramics is possible due to the high pressure phase transformation (HPPT) occurring in the material. The micro laser assisted machining (μ -LAM) system was used to preferentially heat and thermally soften the workpiece material in contact with a diamond cutting tool. In μ -LAM the laser and cutting tool are integrated into a single package, i.e. the laser energy is delivered by a fiber laser to and through a diamond cutting tool. This hybrid

method can potentially increase the critical depth of cut (DoC), i.e., a larger ductile-to-brittle transition (DBT) depth, in ductile regime machining, resulting in a larger material removal rate.

In micro laser assisted drilling (μ -LAD), as is shown schematically in figure 1, the laser beam is transmitted through an optically transparent diamond drill bit and focused precisely at the tool-workpiece interface, where the material is under high pressure induced by the diamond tool. Laser softens the material under the tool which also leads to lower cutting forces and therefore lower tool wear (higher tool life). The focus of this study is on micro drilling of single crystal silicon (100) which is very brittle and hard to drill material by conventional methods. Effects of using laser on process outputs such as edge quality, surface roughness of the wall (of drilled holes) and brittle or ductile mode of the achieved surfaces have been investigated and discussed.

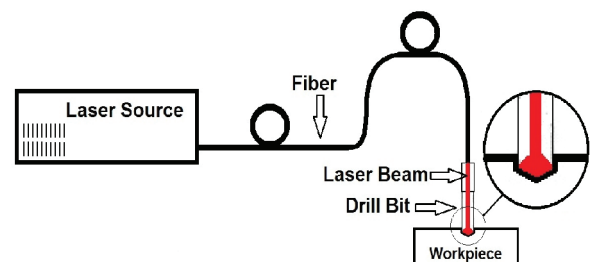


FIGURE 1. Schematic of μ -LAD process

EXPERIMENTAL SETUP

An IR CW fiber laser with wavelength of 1070 nm and max. power of 100 W is used in this investigation. A single edge diamond drilling bit with a 1 mm radius with -45° rake and 5° clearance angle was used for this drilling operation. The Universal Micro-Tribometer (UMT) manufactured by CETR-Bruker Inc. was modified and coupled to the μ -LAD system to

perform all of the drilling tests. Figure 2 shows the experimental setup used for performing the tests. In this setup sample is rotating instead of the tool which is similar to drilling operation in turning process. For this purpose samples are mounted on a precise air bearing spindle. As the sample is rotating, before each test, the tool should be moved to the center of the spindle and sample should be mounted at that position to avoid any inaccuracy. However in this study, which is the first feasibility test of the μ -LAD process, the size of the drilled hole is not the main concern.

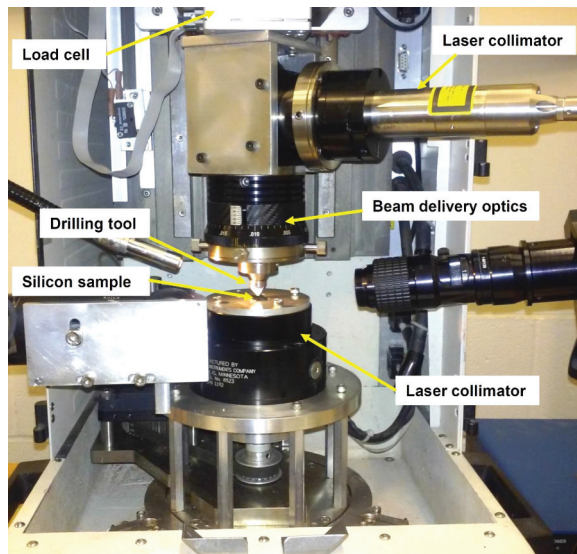


FIGURE 2. μ -LAD setup

Initial results of micro drilling the silicon by a diamond coated drill bit showed that ductile mode cut can be achieved, as it is shown in figure 3, if the feed rate be less than a critical value. This critical value can be determined by increasing infeed rate and check the hole under microscope, after each test, until ductile to brittle mode transition starts. After this value which is mainly depend on unmachined chip thickness and many other parameters such as thrust load or pressure, brittle mode cut starts. The white areas on the surface (wall) of the hole in figure 3 are machined in ductile mode while the dark areas are in brittle mode cut. These dark areas are pits and fractures that occurred during the drilling process which can initiate more cracks and even fail the final part. That is why an ideal hole in a brittle material should have minimum amount of brittle mode cut and depend on application even should be without any imperfection. Despite the face machining (turning) of the brittle materials which is possible

to remove any unwanted surface and subsurface damages by post process techniques such as polishing, drilling is a final process and usually no further process is being used to increase the quality of the product. Therefore using a process with minimal unwanted effects on the material is highly demanded.

EXPERIMENTAL AND RESULTS

Due to low fracture toughness of the brittle materials, fracture happens when they are under tensile stress. Therefore entrance and specially exit of a through hole are very important as tensile stress is much higher than compress stresses in these zones. This experimental study is the first try of using this process for drilling and is focused on benefit of using laser on wall surface roughness and the edge quality of the hole entrance. Therefore a dimple shape hole is drilled in the silicon sample for each test as shown in figure 4 schematically.

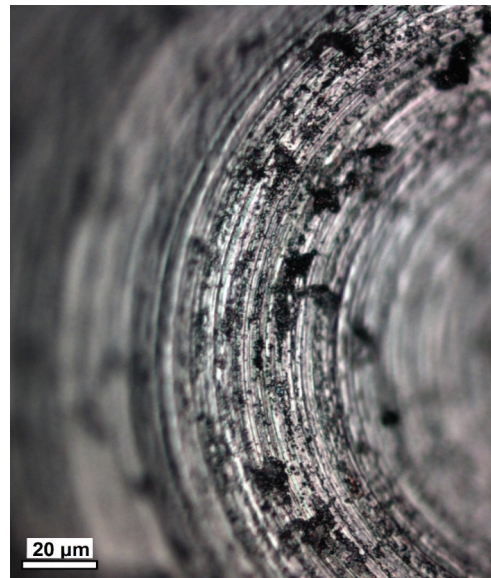


FIGURE 3. Hole wall surface under microscope

Spindle RPM used was 350 as lab scale equipment is used and not rigid as an industrial machine. For each cut, both with laser and with no laser, load was kept about 80 g and tool was fed continuously to reach a 150 micron depth. For the first test, sample drilled with no laser to be able to compare the results later. Figure 5 shows the edge and wall of the hole drilled with no aid of laser. Edge chipping, cracks and brittle mode (in some areas ductile mode) cutting are clearly can be observed. It was possible to infeed the tool more gently, with lower feed rate,

to get a better surface finish, however to show how laser possibly can help to increase the ductility of the material and decrease the damages, sample is drilled in an aggressive manner intentionally.



FIGURE 4. Schematic of a dimple shape hole

Surface roughness parameters of the hole inner surface, R_a and R_z , were measured with a white light interferometer (WYKO). Due to brittle mode cut and the fractures happened during the process, shown in figure 5, obtained surface was rough with R_a of 602 nm and R_z of 5.58 μm . Many cracks at the entrance of the hole, that can propagate later, are due to tensile stresses during the process. For next test a new silicon sample was drilled with aid of laser. It should be mentioned that to keep the condition similar, for both with and no laser tests, same setup used and the only difference was the laser.

TABLE 1. Parameters used for experiments

Parameter	Value
RPM	350
Load	~ 80 g
Laser power	0, 10 W
Depth	150 μ

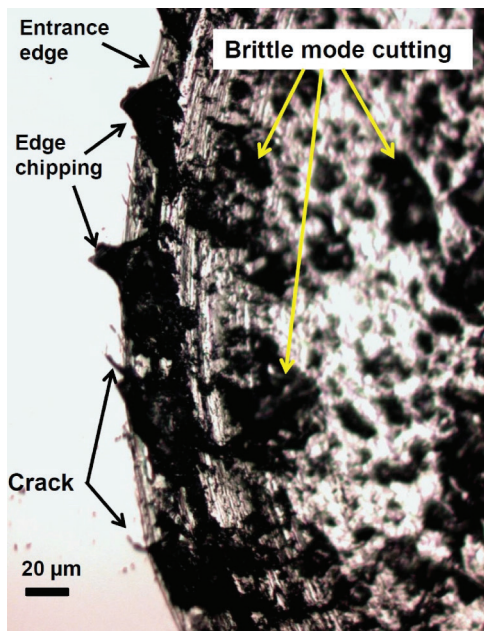


FIGURE 5. Entrance and wall of a hole drilled with no laser

10 W laser power is used for the test with laser, however due to reflection, absorption, error of the laser and scattering the actual laser output was less than 4 W. Low laser power, compare to the power level used in previous works [3-5] on machining, is used as in drilling process the tool respect to the sample is rotating at one spot (in this experiment workpiece is rotating). Test carried out with same condition, same load feed rate and other parameters, as with no laser test except laser. Figure 6 shows the resulting entrance edge and the wall of the hole. Quality of the edge is clearly much better than the hole drilled with no laser (figure 5). Drilling was in ductile mode with almost no sign of fracture. Feed marks also can be seen and surface is obviously smoother than the case with no laser. The R_a and R_z obtained for this surface were 44 nm and 445 nm respectively. No sign of chipping, crack or fracture can be seen on the edge and the wall. In fact laser helped to decrease the brittleness of the material especially at the entrance edge which tensile stress is high.

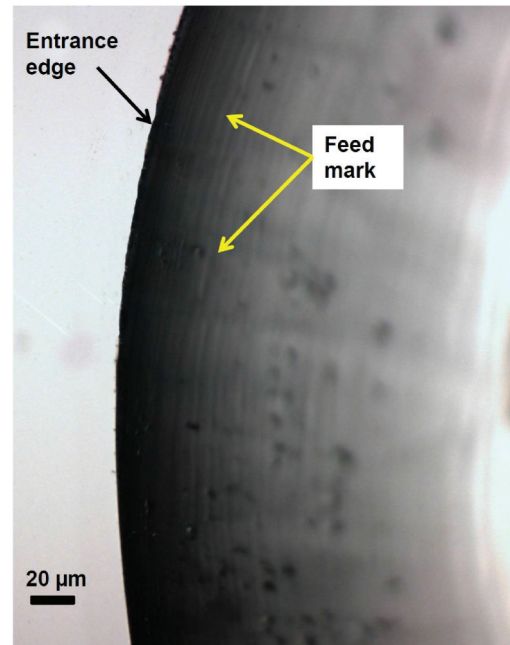


FIGURE 6. Entrance and wall of a hole drilled with laser

To help to visualize the obtained surfaces, a three dimensional (3D) profile of the inner surface of the hole is generated by WYKO profiler for each test. Drilled hole wall with no laser is shown in figure 7. The 3D profile is flattened by software on purpose to be able to see the features on the surface. Blue areas in

figure 7 are the pits caused by brittle mode cut. The 3D profile of the silicon sample drilled with aid of laser, figure 8, shows the feed marks and very minimal imperfections on the surface as well.

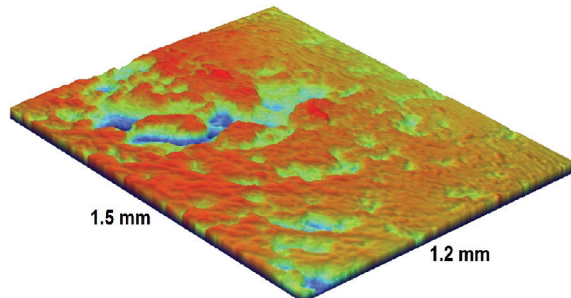


FIGURE 7. 3D image of the inner wall of the hole drilled with no laser

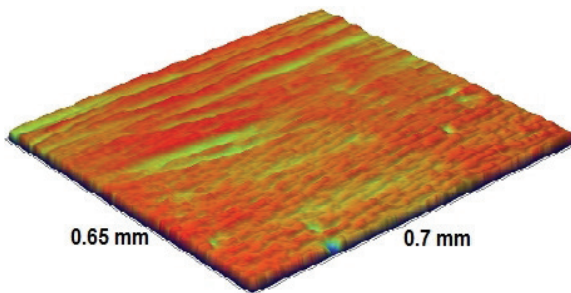


FIGURE 8. 3D image of the inner wall of the hole drilled with μ -LAD process

FUTURE WORK

For future work, μ -LAD process will be used for drilling different materials such as ceramics, carbon fiber reinforced composites, and rocks. Also to understand the effect of other parameters such as RPM, laser power and feed rate more tests are needed to be performed. Studying the tool wear and use of cutting fluid are other possible aspects of the process to be investigated.

CONCLUSION

Results show that the μ -LAD tests were successful in making precise holes on the silicon samples in ductile regime with higher edge quality due to using laser compared to the case drilled with no laser. The μ -LAD system can achieve enhanced ductility, through reduced hardness (and reduced brittleness) resulting from laser assisted heating and thermal softening, to promote more efficient, productive and less costly overall drilling process.

ACKNOWLEDGEMENT

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REFERENCES

- [1] Moon, J. S., Yoon, H. S., Lee, G. B., & Ahn, S. H. (2014). Effect of backstitch tool path on micro-drilling of printed circuit board. *Precision Engineering*, 38(3), 691-696.
- [2] Ahn, S. H., Ryu, S. H., Choi, D. K., & Chu, C. N. (2004). Electro-chemical micro drilling using ultra short pulses. *Precision Engineering*, 28(2), 129-134.
- [3] Ravindra, D., Ghantasala, M. K., & Patten, J. (2012). Ductile mode material removal and high-pressure phase transformation in silicon during micro-laser assisted machining. *Precision engineering*, 36(2), 364-367.
- [4] Mohammadi, H., Poyraz, H. B., Ravindra, D., & Patten, J. A. , Nov. 2014, Micro-Laser Assisted Single Point Diamond Turning On Unpolished Single Crystal Silicon. 29th Annual Meeting of the American Society for Precision Engineering (ASPE), Boston, Massachusetts USA.
- [5] Mohammadi, H., Ravindra, D., Kode, S. K., & Patten, J. A. (2015). Experimental work on micro laser-assisted diamond turning of silicon (111). *Journal of Manufacturing Processes*, 19, 125-128.
- [6] Mohammadi, H., Ravindra, D., & Patten, J. (2013) 'A first investigation of green lasers in micro-laser assisted scratch tests on silicon', In American Society for Precision Engineers (ASPE) 28th Annual Meeting. St. Paul, Minnesota, USA.

STUDY ON IMPROVING MACHINING EFFICIENCY OF INTERNAL MAGNETIC ABRASIVE FINISHING PROCESS

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OVERVIEW OF THE DESIGN

This research proposed a new finishing method for the internal pipe finishing that applied electrolysis to increase machining efficiency. The electrolysis produces aluminum oxide film that modifies the surface morphology that accelerates the planarization process. Then, we remove the film by the magnetic abrasive finishing producing a high finishing surface and minimizing the surface roughness.

In the previous research, the study to polish internal pipe surface utilizing magnetic abrasive finishing was initiated using a slurry mixture. To apply the method in a thicker pipe when the gap increases that caused the magnetic force to fall, a magnetic machining jig was introduced in the design [1]. The design improves push force of the jig towards the internal pipe surface more than ten times stronger. In order to further increase the machining efficiency, a new method that includes electrolysis to speed up the planarization process was proposed. The previous study has confirmed this finishing method feasibility and its capability in reducing finishing time for internal pipe finishing compared to conventional MAF [2]. This method develops oxidation film on the working surface to reduce surface roughness and removal of the film by MAF in a short time. It studied the details of the finishing surface morphology changes that based on roughness value, surface observed through the microscope, and pit structures that formed on the oxidation film. Further study had shown that by maximizing the pit removal, it has shown that a higher surface finishing could be achieved.

The magnetic abrasive finishing with combined electrolysis for internal pipe surface was proposed to produce highly finished pipe internal surface efficiently and environmentally friendly. Figure 1 explains the theory of the finishing process. The electrolysis produces oxidation

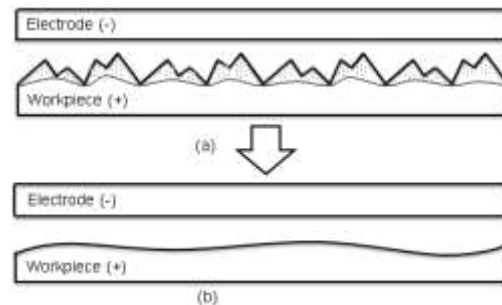


FIGURE 1 Theory of finishing process.

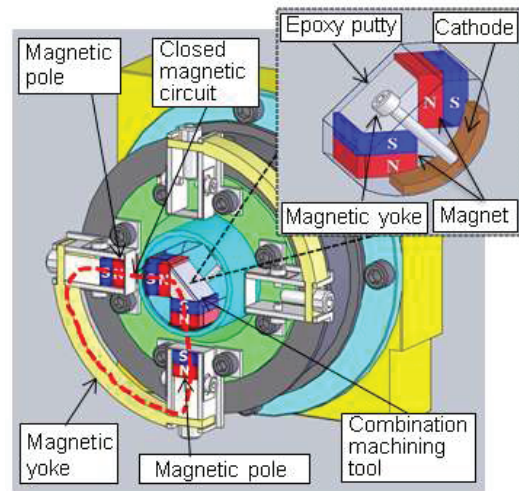


FIGURE 2 Schematic of processing principle.

film on the finishing surface. The process transforms the rough surface that constructed with uneven peaks into a film layer that is porous and easily removed. The two processes; electrolysis, and magnetic abrasive finishing takes place simultaneously, at the same time; with the MAF plays a role to remove the film in a short time. The film removal process by MAF solely does not enough to produce a high finishing surface. Underneath the oxidation film layer, the aluminum surface is still uneven, and this requires a further MAF finishing time to

achieve a high finishing surface as the result shown in Figure 1(b). Overall, this is shorter in total finishing time compared to the conventional method in the study. The finishing time reduction was realized largely due to the creation of the porous structure of the oxidation film and its short time removal.

The two simultaneous processes were realized by the newly designed finishing machine for internal pipe surface as shown in Figure 2. It involves three simultaneous axes movements; the external magnetic poles rotation, workpiece rotation at reverse direction and workpiece stroke. A tool that is capable of magnetic abrasive finishing and electrolysis, which is called combination machining tool is placed in the pipe. The tool is positioned so that its magnet and external poles magnet placed in N-S-N-S sequence. This construction creates a closed magnetic circuit that has more than ten times magnetic force stronger than without it. As the external magnetic poles are rotated using a motor, the combination machining tool will synchronize with its rotation in the pipe. Prior to that, a slurry mixture that contains iron powder, abrasives and polishing agent was adhered to the magnet side of the tool, between the tool and pipe internal wall. The tool rotation movement, pushing towards the wall polishes the internal wall surface.

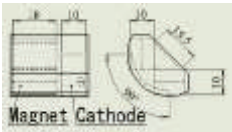
THE RESEARCH SCOPE

The current research scope, views from mechanical perspective, is to investigate the effects of mechanical conditions in regards to proposed new process that influenced the surface morphology changes. In this research, we studied the finishing characteristics of the method. We measured the surface roughness and recorded the photographs of finishing surface to study the morphology change behavior under the modified finishing conditions by using Scanning Electron Microscope (SEM). Notably, during the machining process by MAF, how the surface morphology were affected by different finishing conditions.

THE NEW METHOD ADVANTAGES

The study focuses on three objectives; 1) Finishing to a high finish surface. 2) Reduction of finishing time compared with the conventional method, and 3) The method does not use nor produce acidic or alkali chemical so that it is safe, cheap, easy to handle, and both environmentally and human-friendly.

TABLE 1 Finishing conditions.

Workpiece	A6063 aluminum pipe (Ø40xØ36x150 mm)
Machining tool	Ne-Fe-B rare earth permanent 10x12x18 mm 
Magnetic abrasive mixture	Iron powder 4.0 g (mean diameter 149 µm); WA #10000 0.5 g; Water soluble polishing liquid 2.0 mL
Finishing time	EMAF (2 min) + MAF (12 min)
Pole-pipe gap	8 mm
Pipe rotation speed	200 rpm
Poles rotation speed	50 rpm
Pipe stroke	5 mm/s
Electrode-pipe gap	1 mm
Electrolyte	NaNO ₃ 20% aqueous
Electrolyte amount	2.0 mL
Current Density	0.0025 A/mm ²

The table 1 shows the finishing conditions for the proposed finishing method. To perform the electrolysis and MAF at the same time using the combination tool, the electrolyte sodium nitrate aqueous solution 20% were added to the magnetic slurry mixture 1.0 mL. Another 1.0 mL of the solution is then added directly to the polyester cloth that wrapped the tool, near the electrode side. The electrolyte consists of freely moving ions that allows for electron movement in the electrolytes through the polyester cloth and magnetic slurry mixture.

SURFACE ROUGHNESS AND MATERIAL REMOVAL CHANGES

Figure 3 shows the graph of material removal and surface roughness changes during the finishing process. The surface roughness change has shown significant roughness reduced in the first two minutes of the finishing process. During that period, the formation of oxidation film and removed by MAF mainly contributed to plunge in surface roughness measurement. The intensity in surface roughness reduction reflects the effects of the simultaneous two finishing process. After the two minutes of finishing time, the surface roughness changes become gradually reduced during the MAF single process. The material

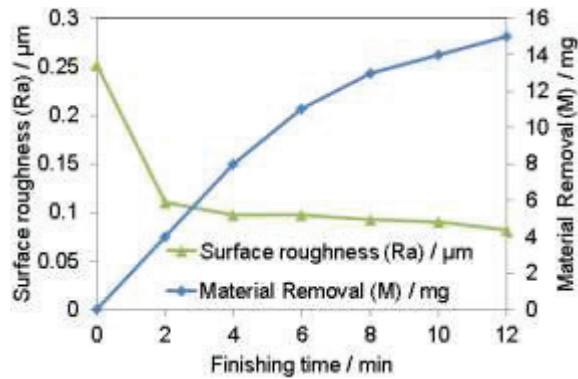


FIGURE 3 Graph of changes of material removal and surface roughness during the finishing process.

removal pattern has shown higher removal rate in the first four minutes, which gradually reduced after that. The removal depth h_r can be calculated by the equation as follow:

$$h_r = \frac{m}{A_p \rho} \quad (1)$$

Where m is the mass, A_p is working area and ρ is material density. Therefore, from the equation above, the material removal depth can be calculated as 4.06 micrometers for the current finishing condition.

EXTERNAL MAGNETIC POLES AND PIPE GAP

Figure 4 shows the photograph of finishing the surface after the processing for different magnetic pole-tube gap. The gap difference translates for magnetic force strength that directly influences the machining force. As observed in Figure 4 (a) gap 4 mm and 4 (b) gap 5 mm, these gap range results suggested too strong machining force for the aluminum pipe, judging from the vertical scratch mark on the surface. The rotation of the tool in the same direction with the marks implies that strong machining force precipitate the scratches. The scratches may be formed during the tool rotation, caused by touching of the tool with pipe surface that breaks through the magnetic particles layer. On the contrary, in Figure 4 (d), (e) and (f) for gap 7, 8 and 9 mm, the horizontal hairlines were clearly seen. These hairlines residual indicated the insufficient machining force or machining time applied. Figure 4 (c) shows the gap 6 mm suggested that the adequate machining force was applied to remove the hairlines with the

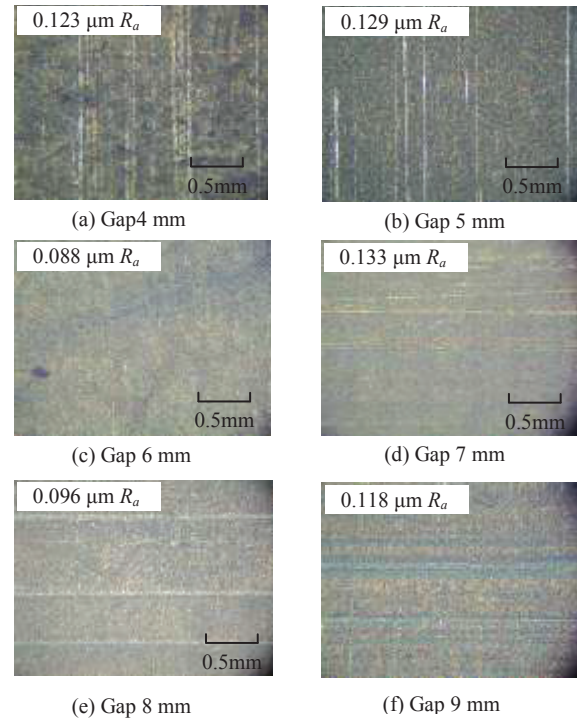


FIGURE 4 Photograph of the finishing surface after processing for different magnetic pole-pipe gap observed under SEM.

force intensity that does not cause visible scratch to the surface.

The test had demonstrated that the right push force by the tool is important in finishing the soft has demonstrated a significant reduction of horizontal lines and the least vertical lines visible, aluminum surface. For stronger force applied, the initial hairlines were removed. This indicated that its effectiveness in removing the hairlines. If the vertical direction scratches could be prevented, for example using a soft material for the tool, there is a possibility for improvement in efficiency.

COMBINATION OF PROCESSING TIME

Figure 5 shows the SEM photograph of the finishing surface for different finishing time combination for simultaneous finishing of MAF and electrolysis. Although fine horizontal lines were removed in all conditions, we could observe that horizontal lines were completely removed for finishing time ratio 2:8 minutes. The initial hairlines were clearly seen for the conditions in Figure 5 (b), (c) and (d) due to incomplete surface removal by MAF. In 5 (a), finishing time total 10 min has demonstrated a reduced finishing processing time.

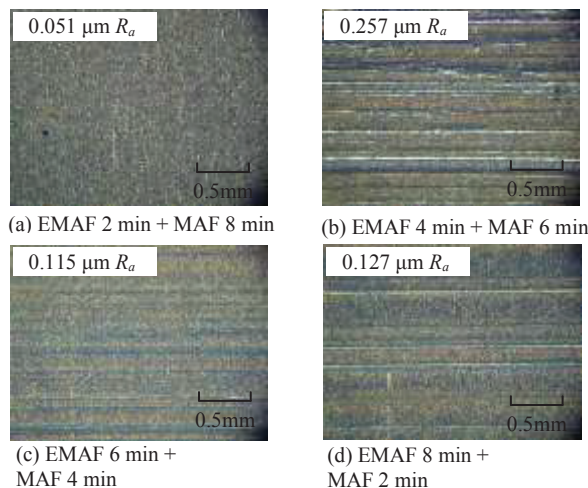


FIGURE 5 Photograph of the finishing surface after processing for different combination of finishing time.

OBSERVATION OF EQUAL PROCESSING TIME OF ELECTROLYSIS AND MAF

Figure 6 shows the SEM photograph of the finishing surface before and after EMAF for 5, 10 and 20 min. We can observe that the planarization takes place swiftly in the early stage, but the surface morphology did not show significant changed after the initial stage. The observation through SEM could not detect significant pit structures in the 5 or 10 min time, but as the oxidation film accumulated, pit structures began to deepen and become visible on the surface for the 20 min finishing time. In line with the theory, the measured surface roughness was gradually increasing after 5 min. This evidence strongly suggested that the increasing of surface roughness could have occurred due to the accumulation of oxidation film residue with prolonged processing time.

Furthermore, it also suggests that the electrolysis speed is higher than the MAF for the experimental conditions. These observations would again stress a significant reason for an additional time for MAF process after the electrolysis is stopped for the removal of the residual oxidation film. Neither MAF nor EMAF process alone is favorable in this study, but the combination of EMAF and MAF at a particular time ratio is critical in producing the high surface finishing in reduced time compared to the conventional method. A more understanding in pit formation and factors such as current value, electrolyte type and materials used could contribute to further improvements of surface roughness and material removal.

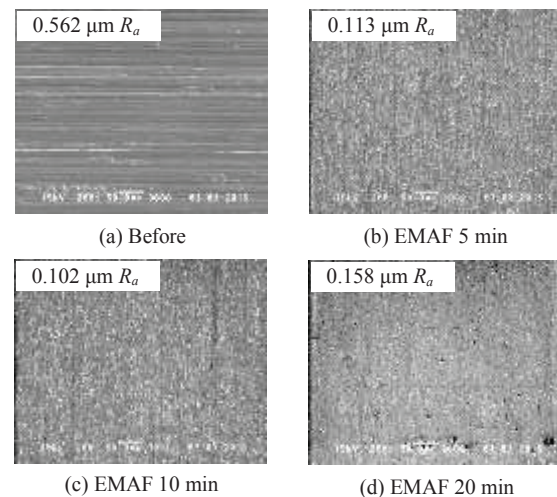


FIGURE 6 Photograph of the finishing surface before and after processing for a prolonged time.

CONCLUSIONS

The proposed method was proved to be effective in reduce finishing time for aluminum pipe finishing. Applications for other materials such as stainless steel, titan and copper are expected to achieve improvement results. The result shown has clearly demonstrated notable features of the finishing characteristics that strongly support the proposed method and its finishing time pattern. Particularly, the need of single MAF at the end of simultaneous finishing period was clarified. The oxidation film formation contributed to speed up the reduction of surface roughness. However, prolonged process causes pit deepening that has a reverse effect, were shown in the test results.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] Y. H. Zou and T. Shinmura, "Study on Internal Magnetic Field Assisted Finishing Process Using a Magnetic Machining Jig," in *Key Engineering Materials*, 2005, vol. 291–292, pp. 281–286.
- [2] M. M. Ridha, Z. Yanhua, and S. Hitoshi, "Development of a New Internal Finishing of Tube by Magnetic Abrasive Finishing Process Combined with Electrochemical Machining," vol. 3, no. 2, pp. 22–29, 2015.

EVALUATION OF THE RANGE PERFORMANCE OF LASER SCANNERS USING NON-PLANAR TARGETS

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INTRODUCTION

The Dimensional Metrology Group (DMG) at the National Institute of Standards and Technology (NIST) is involved in the development of documentary standards for volumetric performance evaluation of laser scanners.

The proposed tests evaluate the performance of laser scanners by determining the measurement error between two derived points[§] at many positions in its work volume. Some of the proposed test positions are along the ranging direction of the laser scanner. Considerable work was done by the ASTM E57.02 committee on “Test methods”, however targets specified by the ASTM E2938 [1] standard are limited to planar targets.

This paper explores non-planar target artifacts such as spherical and trihedral targets, primarily to understand the influence of target geometry on ranging errors.

TARGET SELECTION

Flat planar targets are convenient to use in evaluating the relative range error. These targets are easy to fabricate or obtain commercially. They yield a measurand whose length value is dependent on its angle/orientation, processing method, density and distribution of data collected on the target.

A sphere's derived point is its center and can be calculated with lower uncertainty by using a least-squares minimizing algorithm. For larger distances, larger diameter spheres are needed to capture enough data points on its surface. Since the sphericity (form) of the target sphere affects the determination of its center [2], obtaining commercially manufactured spheres of larger diameters and good sphericity is cost

prohibitive. Spheres measured using laser scanners also suffer from the issue of having increased measurement noise towards the outer periphery of the sphere surface.

Contrast targets used by some laser scanner manufacturers employ proprietary methods to calculate the derived point on the target and hence are difficult to evaluate independently. Further, this derived-point calculation may not use the 3D point coordinates directly, but may use an indirect correspondence between the intensity data and the 3D coordinates.

In some proposed test positions for the volumetric evaluation, the targets may be located asymmetrically (not equidistant from the scanner). In such cases, the derived points calculated from the scans suffer from the problem of unequal range noise and uncertainty for all the four target geometries mentioned in this paper.

MEASUREMENT NOISE VS INCIDENCE ANGLE

A test was conducted to understand the effect of angle of incidence on the range noise of a laser scanner. This test consisted of mounting a flat aluminum plate on an adjustable stage and scanning multiple times at different angles.

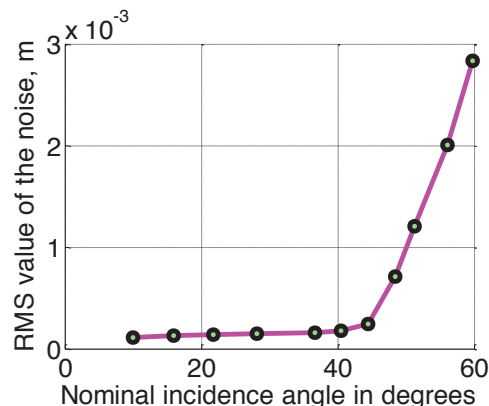


Figure 1. Noise vs incidence angle

[§] A derived point is a uniquely identifiable point on a target that is dependent on its geometry. Examples include sphere center, apex of a pyramid etc.

By fitting a plane to the scan data, the RMS (root-mean-square) values of the residuals (noise) were calculated. It was observed that the noise increases exponentially as the angle of incidence increases beyond 45°.

Figure 1 shows the results of this test. It should be noted that, the instrument manufacturer specifies a value of 50° as the angle beyond which the measurement signal deteriorates resulting in higher noise.

TESTS TO COMPARE RANGE USING TARGETS OF VARYING GEOMETRY

A series of four tests were performed to understand the effect of target geometry on the relative range error (E_{RR}). Each of the four tests varies in either the target geometry, or the procedure to calculate the reference/test length. The setup consists of two locations for the target on a sturdy tripod – a near position and a far position. The reference instrument (RI) was a laser tracker and the instrument under test (IUT) was a laser scanner. The four targets used in these tests were as follows:

- Flat plane target
- Trihedral target
- Hollow sphere target
- Integrating sphere target

For all targets except the integrating sphere target, the RI, IUT and the tripod positions were as illustrated in Figure 2. These three targets were tested simultaneously. For the fourth target (integrating sphere), the RI and IUT were on either side of the target as illustrated in Figure 3.

Of the two positions, the near position was at about 3 m away from IUT and the far position was about 8 m away from the near position, along the line joining IUT and the near position.

To reduce any errors due to the angular encoders, the RI and IUT need to be mounted on the same tripod in succession for taking the reference and IUT measurements. However, that was not the case in the setup described in Figure 2. This setup introduces an error of about 0.03 mm (1σ) in the reference length due to the angular encoder errors which was considerably smaller than the relative range errors.

In all the tests, four derived points were obtained - two at the near position and two at the far position. At the near position, the first derived point was obtained using the RI, and the second derived point was obtained using the IUT. The

target was then moved to the far position and two more derived points were obtained similarly using the RI and IUT. Reasonable care was taken to make sure that the angle/orientation of the targets does not significantly change (with respect to the IUT). This was accomplished by marking the footprints of the tripod on the floor at both the positions and placing them back at approximately the same location during repeat measurements.

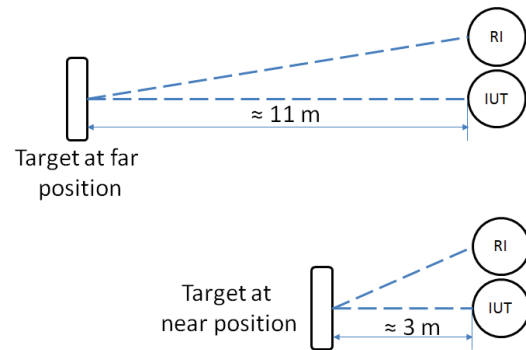


Figure 2. Location of RI, IUT and the two locations of 3 targets in Figure 4.

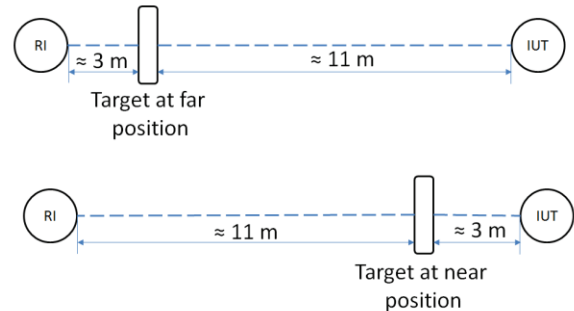


Figure 3. Location of RI, IUT and the two locations of the Integrating sphere target.

The reference length (L_{RI}) was the Euclidean distance between the derived points obtained using the RI at the near and the far positions. The test/IUT length (L_{IUT}) was the Euclidean distance between the derived points obtained using the IUT at the near and the far positions. The relative range error (E_{RR}) was calculated using equation 1.

$$E_{RR} = L_{IUT} - L_{RI} \quad 1$$

In the next few sub-sections, the various target geometries are described and methods to obtain

L_{IUT} and L_{RI} are explained. These values were then used to calculate E_{RR} , using equation 1.

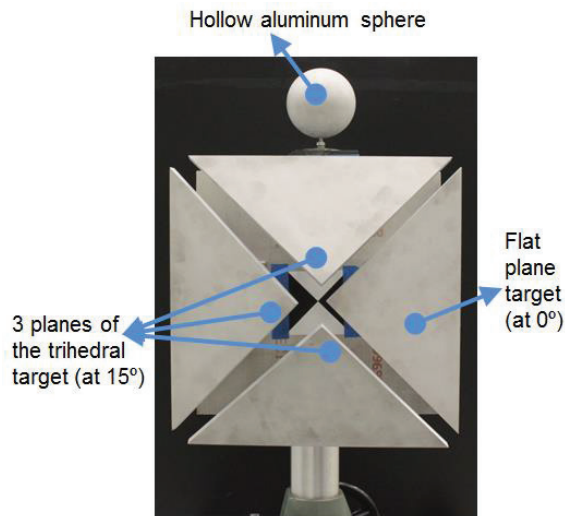


Figure 4. Picture of the trihedral, flat and hollow aluminum sphere targets mounted on a tripod

Flat plane target:

The flat plane target was constructed from a triangular aluminum plate as depicted in Figure 4. Its surface was bead blasted to make its surface diffusely reflecting for the laser scanner. The derived point was the centroid of this plane. This target was mounted on to the tripod and placed at the near position and the far position successively and was measured using the RI and IUT to calculate the relative range error E_{RR} .

Calculating L_{RI} : To obtain the two derived points for the reference length using the RI, seven points were measured on the flat plate at both the positions (near and far) using the SMR (surface mounted retroreflector) walking method [2]. Three points were collected close to the vertices of the triangle, three points at the mid points of the sides and one point at the center of the triangular plate. The Euclidean distance between the two derived points (centroids) at the near and far positions will yield an L_{RI} value. A repeatability test was conducted to estimate the variation in the reference length in the ranging direction and the 1σ variation of 10 measurements was 0.165 mm.

Calculating L_{IUT} : The test length (L_{IUT}) was calculated by using the following procedure:

- a) First the scan data was segmented to obtain the data corresponding to the flat plate.

- b) Then the edge points were eliminated. This was performed by considering a cylindrical region whose axis passes through an initial centroid of the scan data and was along the ranging direction. The radius of this cylindrical region was empirically determined to exclude the edge points. All the points within this cylindrical region were considered for further processing.
- c) A plane fit was performed on the data obtained in step #b) and the residuals were obtained. Points corresponding to the residuals exceeding two times the standard deviation (2σ) value were excluded from this data.
- d) A centroid was calculated for the data obtained in step #c) and this was the derived point for the flat plane.
- e) Another derived point was obtained at the far position and the Euclidean distance between the two derived points was the test length (L_{IUT}).

Note that this procedure for the flat plane target does not adhere to the procedure described in the ASTM E2938 [1] standard.

Trihedral target:

Three aluminum flat triangular plates similar to the flat plane target were used to form a trihedral target and are depicted in Figure 4. The derived point for this target was the intersection of the three planes. This target was first mounted on to the tripod. It was then measured using the RI and IUT while placed at the near position and then at the far position to calculate the relative range error E_{RR} . The three planes of the trihedral target were oriented in such a way that the incident angle of the laser beam from the IUT was $\approx 15^\circ$.

Calculating L_{RI} : To obtain the two derived points for the reference lengths using the RI, the three planes were measured using the method similar to that for the flat plate. Seven points were measured on each of the three surfaces of the target and three planes were fitted to these three sets of data. The intersection of these three planes was the derived point of the trihedral target. The Euclidean distance between the two derived points at the near and far positions will yield L_{RI} . A repeatability test was conducted to estimate the variation in the reference length in the ranging direction and the 1σ variation of 10 measurements was 10 μm .

Calculating L_{IUT} : To obtain the derived point using the IUT, the data on each of the three planes was processed using the steps a), b), and c) described for the flat-plane target. Three planes were then fit to these three sets of data and the intersection of these three planes was the derived point for the trihedral target. Similarly, the second derived point was calculated at the far position and the test length (L_{IUT}) was calculated.

Hollow spherical target:

A bead blasted hollow aluminum sphere (101.6 mm in diameter) was mounted on the tripod as depicted in Figure 4. The derived point for this target was the sphere center.

Calculating L_{RI} : The two derived points for the reference length were the two sphere centers calculated by using the SMR walking method [2]. A set of ≈ 15 points were measured on the sphere surfaces at both the positions using the RI. The sphere centers were then calculated by fitting a sphere using a constrained non-linear least squares algorithm. The Euclidean distance between the two sphere centers at the near and far positions will yield L_{RI} . A repeatability test was conducted to estimate the variation in the reference length in the ranging direction and the 1σ variation of 10 measurements was 0.010 mm.

Calculating L_{IUT} : The two derived points for calculating the test length (L_{IUT}) were obtained by first scanning the spheres at both the positions (near and far) using the IUT. The scan data was segmented to extract the sphere region and was then fitted using a constrained non-linear least squares algorithm.

Integrating sphere target:

A commercial sphere called the “Integrating sphere” was used in this test. This is a truncated sphere (100 mm in diameter) that has a pocket milled into the truncated portion of the sphere. Centered at the bottom of this pocket is a kinematic seat for a 1.5 in. diameter (38.1 mm) SMR/sphere. The concentricity of a 1.5 in. (38.1 mm) diameter sphere and the Integrating sphere, as measured by a coordinate measuring machine (CMM) was $<10 \mu\text{m}$. The derived point for this target was the sphere center.

Calculating L_{RI} : The two derived points for the reference lengths were an average of 20 points as measured by the RI using an SMR when it

was seated in the integrating sphere’s kinematic seat. A set of 10 points were obtained from the RI before the IUT measurement scan and 10 points after the scan. A repeatability test was conducted to estimate the variation in the reference length in the ranging direction and the 1σ variation of 10 measurements was 0.005 mm.

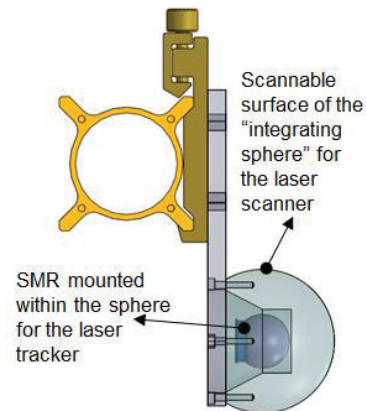


Figure 5. Illustration of the top view of integrating sphere mounted for measurement by the IUT and RI

Calculating L_{IUT} : The two derived points for calculating the test length (L_{IUT}) were obtained by a process similar to that for the hollow sphere. The sphere was scanned at both the positions (near and far) using the IUT. The scan data was then segmented to extract the sphere region and the data was fitted using a constrained non-linear least squares algorithm.

RESULTS AND DISCUSSION

For each of the four targets, eight values of relative range error (E_{RR}) were obtained. Figure 6 shows a plot of the relative range errors for various target geometries and Table 1 shows the corresponding statistics for each target.

A few observations can be made from these results:

- The two spherical targets have the lowest variation (1σ) in the relative range error (E_{RR}) compared to the other targets.
- The flat plane target has the largest variation (1σ) of E_{RR} .
- The E_{RR} values for both the spheres were statistically similar.
- The trihedral target has the lowest absolute mean value of E_{RR} , but has

slightly higher 1σ variation than the spheres.

- The flat plane target has the largest absolute mean value of E_{RR} apart from the largest 1σ variation of E_{RR} . An explanation for a possible reason for such a large bias will be provided later in this section.
- The flat plane targets measure shorter than their reference lengths, whereas the spherical targets measure longer than their reference lengths.

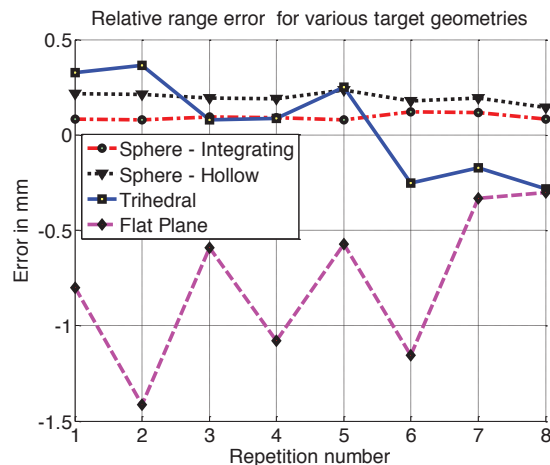


Figure 6. Plot showing the relative range errors for various geometries

Table 1. Statistics for the relative range error tests on four targets

Target type	Nom. length (L_{NOM})	1σ of L_{RI} (mm)	Mean of E_{RR} (mm)	1σ of E_{RR} (mm)
Sphere-I**	7.8 m	0.005	0.094	0.017
Sphere-H**	8.4 m	0.010	0.195	0.028
Trihedral	8.4 m	0.010	0.051	0.259
Flat plane	8.4 m	0.165	-0.781	0.403

Flat plane targets using the centroid as a derived point resulted in relative range errors that were larger compared to other targets used in these tests. The measurement of the derived-point to derived-point lengths for data from both the instruments relies on the target plates to be perfectly parallel.

For two perfectly parallel planes, if the centroids determined by RI and IUT are not identical, then the error is negligible as this error is not in the

ranging/sensitive direction (cosine/second order error). However, if the planes are not parallel, an error in the centroid determination will result in an error in the sensitive direction (first order error). This was exacerbated by the fact that the reference length was calculated based on a centroid determined by only seven points at each position.

As an example, consider two flat plane targets at the near position and the far position that are not parallel and are at 1° with respect to each other. Also consider that there is a 5 mm error in locating the centroid in the plane perpendicular to the sensitive direction. Such a setup will result in an error of $87 \mu\text{m}$ error in the relative range.

We plan to perform more tests using the flat plane target by making sure that the targets at the near and far positions are truly parallel and using alternative processing methods. The parallelism might be achieved by designing a kinematic mounting setup that uses a plane mirror and a laser for alignment.

One alternative processing method is to fit the data from the near and far locations using a parallel plane fitting algorithm [3] and calculating the distance between the fitted parallel planes. This method can still suffer from a first order effect with respect to the parallelism of the planes. Other methods to try include a bounding box method and the intersection of diagonals [1].

CONCLUSION

Four targets were tested for measuring relative range error of a laser scanner. Of these, one used a flat plane target. The flat plane target, where the centroid was the derived-point for the reference length results in larger relative range errors compared to the other three targets.

In contrast, spheres and trihedral targets have lower relative range errors and are less sensitive to their angle/orientation with respect to RI and IUT.

These tests indicate that the target geometry, alignment and the processing method affect the relative range error. More tests are planned (various lengths and processing methods) to understand the effect of target geometry on the relative range error.

** Sphere-H is the hollow sphere and Sphere-I is the integrating sphere

REFERENCES

- [1] ASTM E2938 standard: "Test method to evaluate the relative-range measurement performance of 3D imaging systems in the medium range"
- [2] Rachakonda, P., Muralikrishnan, B., Lee, V., Sawyer, D., Phillips, S., Palmateer, J., "A Method of Determining Sphere Center to Center Distance Using Laser Trackers For Evaluating Laser Scanners", Proceedings of the American Society for Precision Engineering, Annual Conference, Boston, Massachusetts, November 09-14, 2014.
- [3] Shakarji, C., Srinivasan, V., "Theory and algorithms for weighted total least-squares fitting of lines, planes, and parallel planes to support tolerancing standards", Journal of Computing and Information Science in Engineering, 13(3)

MICRO-LASER ASSISTED MACHINING: AN INNOVATIVE TECHNOLOGY TO MACHINE OPTICS PRODUCTIVELY

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ABSTRACT

A wide range of materials including metals and alloys, ceramics, glasses, semiconductors and composites are manufactured to meet service requirements to a given geometry, accuracy, finish and surface integrity. Metals and alloys in general are easier to machine in the ductile regime because of their high fracture toughness, low hardness, non-directional bonding, low porosity, large strain to fracture and high impact energy. Non-metals, on the other hand, such as ceramics and semiconductors, are characterized by covalent or ionic bonding, limited slip systems for plastic deformation, high hardness and low fracture toughness. It is due to these major differences that ceramics and semiconductors are classified as “difficult to machine” nominally brittle materials.

In this paper an innovative and novel technique termed micro-laser assisted machining (μ -LAM) will be analyzed for machining optical materials such as Silicon (Si), Calcium Fluoride (CaF_2) and Zinc Sulfide (ZnS). The goal of this study is to evaluate the effect of preferential laser heating during the material removal process.

INTRODUCTION

Ceramics are hard, strong, inert and lightweight. They also have good optical properties, wide energy bandgap and high maximum current density. This combination of properties makes them ideal candidates for the optic and optoelectronic industries. Manufacturing these materials without causing surface and subsurface damage is extremely challenging due to their high hardness, brittle characteristics and poor machinability. Current limitations for machining optical ceramics include the high cost of processing and low product reliability. The cost is mainly due to expensive tools that wear

out rapidly, long machining time, low production rate and the manufacturing of satisfactory surface roughness, figure and form.

Ceramics and semiconductors however, have some attractive properties compared to metals and polymers, which make them useful for specific applications such as for high temperature and high power devices. Ceramics are a very broad class of materials with a wide range of properties. [1] One often distinguishes between traditional ceramics such as tableware, pottery, sanitary ware, tiles, bricks and advanced engineered ceramics, where depending on the specific application one or several of the following properties are utilized: high strength at high temperatures, wear resistance, corrosion resistance, low density, low thermal conductivity, low electric conductivity, favorable optical, electrical or magnetic properties, and biological compatibility.

Past research has demonstrated that ductile regime machining of these materials is possible due to the high pressure phase transformation (HPPT) occurring in the material caused by the high compressive and shear stresses induced by the single point diamond tool tip. [2],[3] To further augment the ductile response of these materials, traditional diamond turning is coupled with a novel micro-laser assisted machining (μ -LAM) technique.

The patented μ -LAM technology directly heats and thermally softens the workpiece material, in the chip deformation and generation zone, increasing the material's ductility. The fundamental concept of this process is illustrated in Figure 1. Improved ductility that results from reduced material hardness allows for easier chip formation, decreased brittleness and ultimately higher material removal rates, i.e. better tool performance and increased productivity, which leads to lower manufacturing costs.

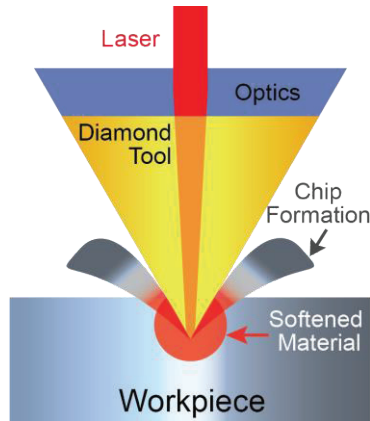


FIGURE 1. Schematic of the μ -LAM process.

The focus of this paper is to evaluate the machinability and surface finish of optical materials via the μ -LAM process. Machining parameters such as the depth of cut, cross feed, cutting speed and laser power are optimized in order to make the manufacturing process more efficient. Materials that will be discussed in this paper include optical grade Silicon (Si), Calcium Fluoride (CaF_2) and Zinc Sulfide (ZnS).

EXPERIMENTAL APPROACH

All machining tests were performed on an AMETEK Precitech N250 Ultra diamond turning lathe with a 4.3 nm positional accuracy. The N250 diamond turning lathe is designed for the production of optical lenses, optical mold inserts and mirrors as well as small precision mechanical components. Figure 2 shows the μ -LAM system mounted and setup on the N250 to machine optical Si. The μ -LAM system replaces the existing tool-post where it is a simple bolt-on system that takes less than 10 minutes to retrofit.

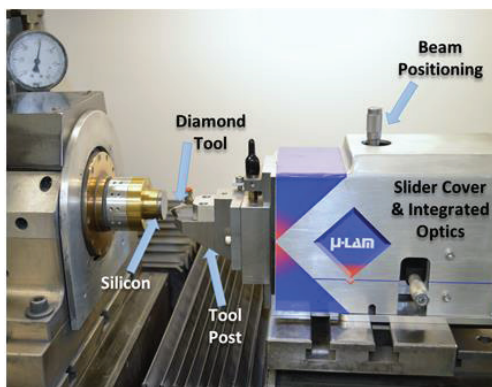


FIGURE 2: The μ -LAM system retrofitted on to the AMETEK Precitech N250 Ultra.

The μ -LAM system is coupled to a 1064nm YAG laser via a fiber optic cable and collimated lens. The diamond tool is optically transparent to the wavelength of the laser and fabricated to perform as a focusing lens that directs the beam to the tool's cutting edge radius. Figure 3 shows the configuration and components of the μ -LAM system.

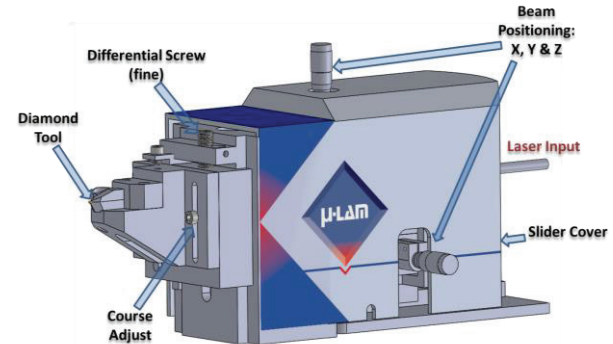


FIGURE 3: The μ -LAM system configuration: tool post, integrated beam delivery optics and collimated laser source.

A total of three machining passes were performed on each materials: roughing, semi finishing and finishing. Only the finishing passes will be discussed in this paper as it is most relevant for the final surface finish of the machined part. Table 1 shows the machining parameters used for the finishing pass.

Table 1: μ -LAM parameters for finishing pass

	RPM	Feed ($\mu\text{m}/\text{rev}$)	Depth (μm)	Laser (W)
Si	10k	1	5	10
CaF_2	8k	1.5	2	5
ZnS	8k	1	2	5

The parameters given in Table 1 are optimized based on the single crystal diamond tool geometry (1 mm nose radius, -25° rake and 7.5° clearance) and machining setup. The ability to fabricate single crystal diamond tools with extremely smooth finish ($R_a \approx 3 \text{ nm}$), makes it ideal for diamond turning optical materials to improve surface finish with a high degree of accuracy.[4] Synthetic single crystal diamond tools with approximately the same properties as natural diamond, are used for all experiments discussed in this study. A range of μ -LAM

parameters (shown in Table 2) were tested in order to choose the optimal conditions.

TABLE 2: Range of parameters tested for optimization

RPM	Feed ($\mu\text{m}/\text{rev}$)	Depth (μm)	Laser (W)
2k – 10k	0.3 - 2	0.5 - 10	2 - 30

All machining tests were performed with odorless mineral spirits (OMS) to stay consistent with conventional diamond turning practices.

RESULTS SUMMARY AND DISCUSSION

The laser assisted machined samples were compared to conventional diamond turned samples. Both tests (with and without laser) were kept consistent to keep only the laser heating as the variable. This will enable the evaluation of the thermal softening effect during the material removal process.

The machined surface roughness was measured using a Taylor Hobson white light interferometer with $<1\text{\AA}$ resolution. An ISO 25178 standard 0.08 mm Gaussian filter was applied for all surface roughness measurements reported in this paper. A sample measurement profile of Si machined using the μ -LAM system is shown in Figure 4.

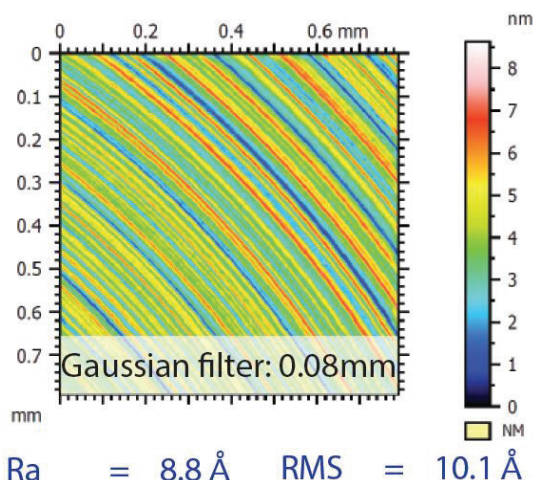


FIGURE 4: A white light interferometric profile of a laser assisted diamond turned Si surface.

Table 3 compares the surface roughness (RMS) of the machines surfaces using the μ -LAM technique with conventional diamond turning.

TABLE 3: Surface roughness (RMS) comparison of machined samples

	μ -LAM	Conventional*
Si	10 $\text{\AA}$	42 $\text{\AA}$
CaF ₂	15 $\text{\AA}$	66 $\text{\AA}$
ZnS	18 $\text{\AA}$	35 $\text{\AA}$

* measurements we performed on cosmetically imperfect areas

It is important to note that for all conventionally diamond turned samples, visible signs of aesthetic imperfections were present in the form of spokes or hazy patches. However samples machined with the μ -LAM system did not show signs of any aesthetic imperfection. High magnification optical microscopy was performed on all machining surfaces to evaluate surface imperfections such as voids, cracks, pits, etc. Figure 5 compares optical microscope images of machined Si using the μ -LAM system and conventional diamond turning.

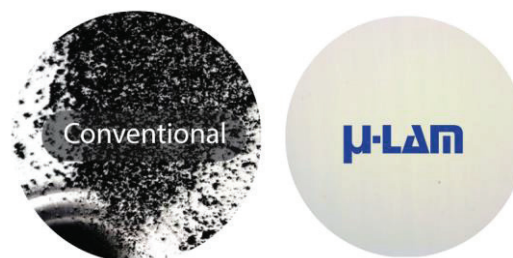


FIGURE 5: Optical microscope surface images of Si with and without the μ -LAM system. For the conventionally turned image (left), the area where a spoke pattern exist has been imaged and shown.

Typically in conventional machining there are several factors that could affect the surface quality of the workpiece during machining, including tool geometry, tool wear/damage, external vibration (from the tool assembly or spindle), cutting speed, feed, depth of cut, cutting fluids and friction forces. However, with the assistance of laser radiation, the workpiece material becomes softer, more ductile and thus more machinable resulting in a more continuous chip flow. Previous tests have shown that a

more continuous chip formation process during material removal is a result of plastic deformation (versus fracturing the material).[5] In order to obtain optical quality surface finishes, material removal via fracture occurrence, also termed brittle mode machining, must be avoided.

An ongoing challenge when machining crystalline optical materials is its varying behavior with respect to its crystal plane. This means that the material behaves differently at different areas of the same substrate, i.e. 90°, 180°, 270° etc. The reason for this is due to its varying critical depth or ductile-to-brittle transitions (DBT) depths (brittleness measure) along each plane on the same substrate. The varying machinability planes often cause spoke patterns, hazy patches and other aesthetic imperfections on the machined surface. Often times this is removed by post processing such as polishing that at times could be the most costly phase in the manufacturing process. Any depth exceeding the critical depth, which is also known as the DBT depth, will exhibit brittle fracture. The extent of brittle fracture ranges from the onset of fracture (at the DBT zone) to a fully fractured condition, as the depth is increased beyond the DBT depth.

It has been previously demonstrated that the μ -LAM technique increases the DBT depths due to the preferential heating process.[6] In other words, the test results suggest that micro laser-assisted diamond turning are successful in enhancing the ductile response of nominally brittle materials. This is primarily due to the laser absorption in both the high pressure phase transformed and atmospheric phases. The laser heating is focused along three tool paths: the leading edge, directly beneath the tool (primary emphasis) and trailing edge.

As the tool tip induces a significant amount of pressure on the workpiece (>10GPa) during machining, it is common that subsurface damage in the form of metastable and amorphous phases are often present after machining. However, previous μ -LAM tests on Si showed signs of healing or recrystallization where no subsurface damage was present after the machining process.[7] The recrystallization of the metastable and amorphous phases to the original crystalline phase is yet another an

added benefit of the μ -LAM process. This also ensures that the material will remain its mechanical, optical and electrical properties after the machining process requiring no further post processing.

CONCLUSION

Single point diamond turning (SPDT) was chosen as the preferred material removal method as it offers better accuracy, quicker fabrication time and lower cost when compared to grinding and polishing.[8] The high demand for quality surfaces in optical and electronic industry has been consistently pushing and breaking the barriers in various nanotechnology areas such as SPDT. Such advances are essential in order to economically produce high quality semiconductor, ceramic and glass parts. SPDT is well known for providing mechanical and optical advantages due to the level of precision of the equipment and the diamond cutting tool.

Micro laser-assisted machining was successfully demonstrated as evidence by the significant improvement in surface finish: roughness and aesthetics. The chart in Figure 6 shows that the surface roughness (RMS) improved over 75% for Si and CaF₂ and approximately by 50% for ZnS.

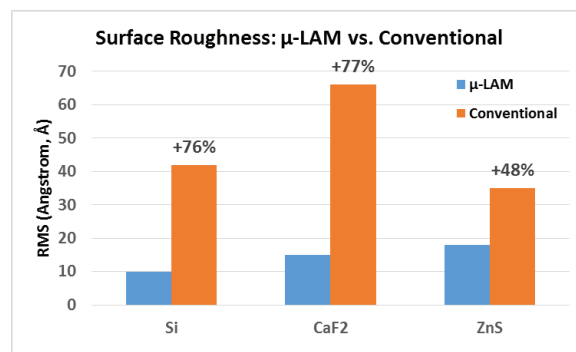


FIGURE 6: Chart comparing surface roughness of μ -LAM and conventionally machined surfaces.

The μ -LAM technique was successful in demonstrating the enhanced thermal softening in Si, CaF₂ and ZnS resulting in a more ductile-like form of material. The results from this study also will benefit the manufacture of brittle materials as laser heating is proven to decrease the brittle response in ceramics and

semiconductors, which can result in high productivity rates (i.e. higher material removal rates).

ACKNOWLEDGEMENTS

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REFERENCES

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- [1] D.W. Richerson, 1992, "modern Ceramic Engineering: Properties, Processing and Use in Design", 2nd ed., Marcel Dekker, New York.
 - [2] Mohammadi, H., Poyraz, H. B., Ravindra, D., & Patten, J. A. , Nov. 2014, Micro-Laser Assisted Single Point Diamond Turning On Unpolished Single Crystal Silicon. 29th Annual Meeting of the American Society for Precision Engineering (ASPE), Boston, Massachusetts USA (Published).
 - [3] Deepak Ravindra and John Patten, "Micro-Laser Assisted Single Point Diamond Turning Feasibility Tests of Single Crystal Silicon", American Society for Precision Engineers (ASPE) 28th Annual Meeting, St. Paul, Minnesota, October 2013.
 - [4] J. Wilks, 1980, "Performance of Diamonds as Cutting Tools for Precision Machining", Journal of Precision Engineering, v 2 (2), pp. 57-72
 - [5] Deepak Ravindra and John Patten, "Ductile Regime Single Point Diamond Turning of Quartz Resulting in an Improved and Damage-Free Surface", Journal of Machining Science and Technology, Vol. 15, No. 4, pp. 357-375, 2011.
 - [6] Deepak Ravindra and John A. Patten, "Micro-Laser Assisted Machining: The Future of Manufacturing Ceramics and Semiconductors" Journal of Sensors and Materials, Special Issue, v26, Number 6, 2014, pp. 417-427.

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- [7] Deepak Ravindra, Muralidhar K. Ghantasala and John Patten, 2011, "Ductile Mode Material Removal and High-Pressure Phase Transformation in Silicon during Micro-Laser Assisted Machining" Journal of Precision Engineering (ASPE), Vol. 36, Issue 2, pp. 364-367.
 - [8] FZ Fang, XD Liu, LC Lee, 2003, "Micro-machining of Optical Glasses- A Review of Diamond- Cutting Glasses", Indian Academy of Sciences, Vol 28, Part 5.

Balancing concepts for high speed aerostatic spindles

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With the availability of high speed air-bearing spindles, high performance cutting (HPC) has become a field of interest in ultra-precision machining. As the centrifugal force increases with the square of the number of revolutions per minute, high spindle speeds require excellent balancing of the rotor [1]. Due to the comparatively low stiffness of air-bearings, unbalances affect the achievable surface quality, especially at higher spindle speeds [2].

The general principle of balancing involves the addition, removal or rotary redistribution of masses on the spindle rotor to generate defined centrifugal force changes which counteract the residual unbalance, as shown in Figure 1.

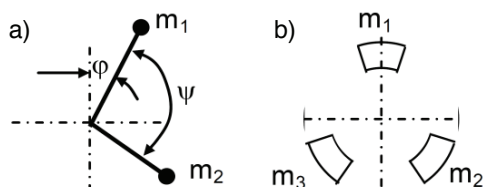


FIGURE 1. a: Rotary redistribution (ψ, φ) of mass (m_1, m_2) and b: addition or removal of mass (m_1, m_2, m_3) on defined circumference positions for balancing the rotor.

In ultraprecision machining, a widely applied method to eliminate unbalances is the addition (or removal) of set screws in circumferentially distributed threaded holes on the workpiece or fly-cutter. The disadvantages are the manual, time-consuming procedure and the low achievable adjustment accuracy, as the number of available set screw weights and the possibility to install additional, smaller counterweights is limited. The required mass adjustment accuracy down to a few micrograms or below cannot be achieved by this method.

In this paper, two novel balancing concepts, which are based on the principles of mass removal and a rotary redistribution of mass, are investigated.

Micropumps

The first balancing concept involves the defined removal of a fluid from a chambered balancing disk (according to Figure 1b). In comparison to the conventional set screw balancing, this balancing concept can be operated by microcontrollers, allowing for an automated balancing process at operating speed while providing the desired amounts of mass without intervention of the machine operator.

Another advantage is the possibility of generating very low centrifugal force changes by releasing defined amounts of fluid in the range of few microliters ($1 \mu\text{l}$ water $\approx 1 \text{ mg}$). Therefore, two commercially available microfluidic pumps, a piezoelectric membrane pump and a peristaltic pump, were tested for their capabilities of delivering defined minimum fluid volumes of water.

The flow rates of the pumps can be set by the variation of the voltage amplitude and stroke frequency (for the piezo pump) or the pulse-width modulation (PWM) signal (for the peristaltic pump). Generally, the flow rate is proportional to these parameters.

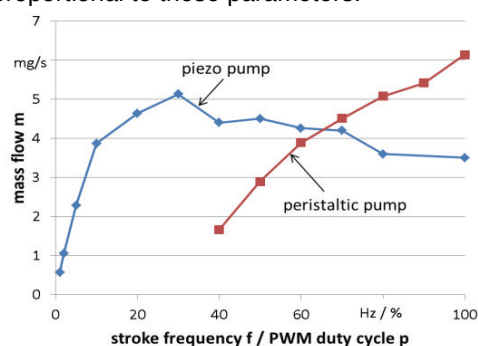


FIGURE 2. Generated flow rates according to the stroke frequency/PWM duty cycle for each pump presented in mg/s for comparability.

In the executed tests with the piezo pump, the amplitude is being held constant at 100 V and the frequency is varied between 0 and 100 Hz in predefined intervals. For the peristaltic pump,

the minimum PWM signal to drive the pump is 40%, the variation includes 10%-steps up to a PWM signal of 100%.

The results for the piezo pump show that the pumped volume per stroke declines with increasing frequency, as seen in Figure 2. This is expected to be caused by a low-dynamic behavior of the piezo controller. At the peak frequency of 100 Hz, a volume flow equal of 0.035 μl per pumping stroke (equal to 0.035 mg) was observed.

For the peristaltic pump, a mass flow between approx. 1.6 mg/s and 6 mg/s was calculated (see Figure 2). However, the minimum pumpable fluid amount is limited by the working principle of the pump, which involves a periodical compression and release of a flexible tube with an eccentric. Thus, the amount of fluid released during one revolution is nearly constant.

In total, both pumps are capable of delivering defined low amounts of fluid. However, the capabilities of the piezo pump are considered more suitable for balancing operations on high speed spindles due to the very small achievable fluid amounts per pump stroke.

Ultrasonic Motor

In addition to the fluid based balancing system, an ultrasonic motor based system was investigated. In contrast to the fluid system this approach uses rotatable masses to influence the balancing state of the spindle rotor (c.f. Figure 1a). The system consists of two ultrasonic traveling wave motors (USM), each driving an unbalance ring. The two rings have different diameters but equal unbalance (17 g·mm). The USMs are mounted facing each other so that both unbalance rings are in the same plane (see Figure 3).

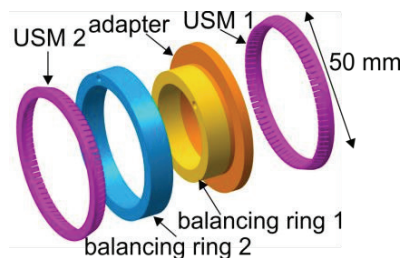


FIGURE 3. Exploded drawing of the two USM with the corresponding balancing rings and an adapter.

This arrangement allows for the system to create unbalances from 0 to 34 g·mm in arbitrary radial direction.

The rotational step width of the unbalance rings mainly depends on the achievable angular accuracy of the USM. In order to rotate the USM, it has to be excited with about 31,000 pulses per second. After a startup phase it reaches a steady speed. In order to create a small step size, only the startup phase is used to create the rotating motion. In Figure 4, multiple pulses and the resulting rotation angles are shown. The achievable minimum angular step size is 0.009°. The resulting minimal balancing step of this setup is then 90 $\mu\text{g}\cdot\text{mm}$.

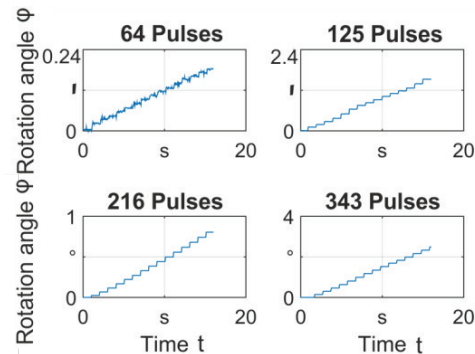


FIGURE 4. Correlation between pulse size and rotation step angle of the USM.

Conclusion

Both presented concepts show promising results regarding the achievable accuracy of mass removal/ mass shift in a static setup. In order to realize high precision balancing on high speed air bearing spindles, further efforts have to be made in order to determine the capabilities of the introduced concepts in a rotating setup.

Acknowledgements

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References

- [1] Dyer, S. W.; Al-Regib, E. I.: Effects of active balancing on high-speed milling precision. In: Proceedings of the ASPE – Annual Meeting, Monterey, CA, USA, 1999.
- [2] Knapp, B.; Arneson, D.; Oss, D.; Liebers, M.; Vallance, R.; Marsh, E.: The Importance of Spindle Balancing for the Machining of Freeform Optics. ASPE Spring Topical Meeting, pp. 74-78, Charlotte, NC, USA, 2011.

UNCERTAINTY OF IN-PROCESS ROUNDNESS MEASUREMENT USING THE MULTI-PROBE ERROR SEPARATION METHOD

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INTRODUCTION

The multi-probe error separation method has been used in spindle research, with emphasis on estimation of spindle error motions. The multi-probe error separation method utilizes multiple sensors arranged around the circumference of a rotating artifact. The method can use an un-calibrated, but nominally round, artifact as the target; and separates the spindle motion errors from the roundness errors of the artifact. Most researchers using this method have been focused on the spindle error motions. In this work, we investigate the use of the multi-probe error separation method as a means of in-process measurement of workpiece roundness during machining and grinding operations.

Uncertainty analysis of error separation techniques has been limited, with more work done on the multi-step method [1]. Recently some researchers have investigated the uncertainty sources in the multi-probe error separation method, using various approaches [2] [3] [4]. However, that work is more concerned with uncertainty in the characterization of spindle error motions.

This research investigates the uncertainty of roundness measurements obtained using the multi-probe error separation method; and carried out in a shop-floor environment, with emphasis on large-scale workpieces. The roundness errors of the workpiece from the multi-probe method are compared to a measurement made by a high-precision coordinate measuring machine (CMM). Sources of uncertainty in the method are identified; and an uncertainty budget with the sources, values, distributions, divisor (factors) and expanded uncertainty is detailed.

MULTI-PROBE ERROR SEPARATION

Background

The multi-probe error separation method uses three displacement probes (e.g. capacitance gages) arranged around a nominally round,

rotating artifact. The probes simultaneously measure the spindle error motion combined with the workpiece roundness error [5]. The capacitance gages are arranged around the circumference of the workpiece. The capacitance gages are labeled as $M_A(\theta)$, $M_B(\theta)$ and $M_C(\theta)$. $M_A(\theta)$ is along the x direction, $M_B(\theta)$ is referenced by angle β and $M_C(\theta)$ is referenced by angle γ . θ is the angle of rotation the workpiece follows. This can be seen in Figure 1.

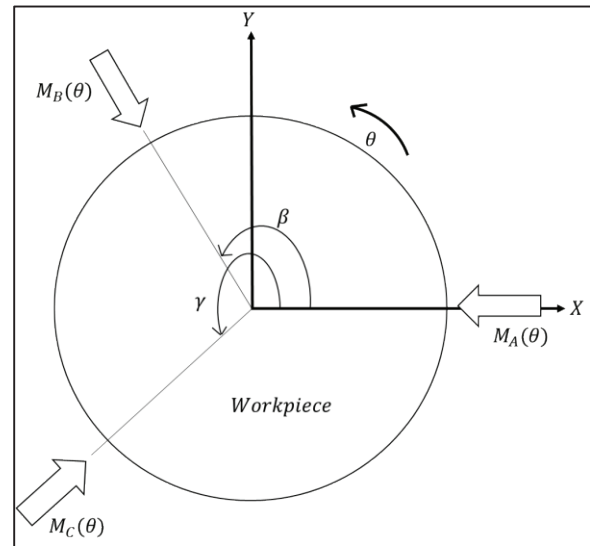


FIGURE 1. Multi-probe error separation, with three probes, for workpiece roundness and spindle error motion measurements.

The equations for the measurements are a combination of the part roundness, $P(\theta)$, including the phase shift of the measurement point locations, and the $x(\theta)$ and $y(\theta)$ of the spindle part error motions [5]:

$$M_A(\theta) = P(\theta) + x(\theta) \quad (1)$$

$$M_B(\theta) = P(\theta - \beta) + x(\theta)\cos(\beta) + y(\theta)\sin(\beta) \quad (2)$$

$$M_c(\theta) = P(\theta - \gamma) + x(\theta) \cos(\gamma) + y(\theta) \sin(\gamma) \quad (3)$$

A weighted sum, $M(\theta)$, of these three measurements is created using coefficients a and b , which are functions of the angles β and γ .

$$M(\theta) = M_A(\theta) + aM_B(\theta) + bM_c(\theta) \quad (4)$$

The roundness of the workpiece can be expressed as a Fourier series, with the Fourier coefficients A_k and B_k ; where k is the number of undulations per revolution (UPR). The roundness equation, is:

$$P(\theta) = \sum_{k=1}^{\infty} A_k \cos k\theta + B_k \sin k\theta \quad (5)$$

Eq. (4) can also be expressed as a Fourier series.

$$M(\theta) = \sum_{k=1}^{\infty} F_k \cos k\theta + G_k \sin k\theta \quad (6)$$

Equating Eq. (4) and Eq. (6), the following matrix can be solved for A_k and B_k .

$$\begin{bmatrix} \beta_k & \gamma_k \\ -\gamma_k & \beta_k \end{bmatrix} \begin{Bmatrix} A_k \\ B_k \end{Bmatrix} = \begin{Bmatrix} F_k \\ G_k \end{Bmatrix} \quad (7)$$

With A_k and B_k now known, the workpiece roundness, $P(\theta)$ using Eq. (5), can be determined.

Uncertainty Analysis

Uncertainty analysis consists of determining the sources of uncertainty, a value, distributions, the distribution divisor (factor) and the expanded uncertainty. Methods from the *Guide to the Expression of Uncertainty in Measurement* [6] and *ASME B89.3.4* [7], will be used to determine the uncertainty budget. All uncertainty sources in this paper are assumed non-correlated.

ASME B89.3.4 [7] has a non-mandatory appendix, which describes the procedure for evaluating uncertainty of axis of rotation measurements. This standard is oriented toward evaluation of spindle error motions, but uncertainty sources that contribute to spindle error measurements will also contribute to the workpiece roundness. Additional error sources arise in this research because the measurements are intended to be performed on the shop-floor.

The effect of the sensor angle locations for the multi-probe method have been looked at extensively for uncertainty analysis, since Eq. (7) can become ill-conditioned and harmonics can be suppressed for some sensor angle combinations, resulting in incorrect results.

Other error sources considered in this paper are, probe sensitivity, including resolution and accuracy, probe plane misalignment due to use of large size workpieces and the resulting large distance between probes, noise in the probes which include thermal drift and structural vibration, sensitivity of the probes not normal to the surface of the workpiece, and repeatability of the measurement.

EXPERIMENTAL SETUP

Experiments were conducted on a $\frac{3}{8}$ " thick steel plate, with a diameter of approximately 18". The steel plate is bolted onto a 4" diameter steel cylinder and mounted in the spindle of an Okuma LC-20 lathe. The setup can be seen in Figure 2.

The measuring fixture is made of the same $\frac{3}{8}$ " thick steel plate. It is a half-ring design, with an inside diameter of 20.5" and an outside diameter of 21.5". It has threaded holes to accommodate the L-shaped brackets which hold the capacitance gages at the selected angles (0° , 31.9115° and 127.1043°). These angles were selected from a simulation of the multi-probe error separation method and simulated noise expected in a shop-floor setting. The L-brackets are slotted so they can slide radially to accommodate a range of disk diameters. The fixture also holds an optical sensor that senses the passage of a small piece of retro-reflective tape on the workpiece, and whose output is used to clock workpiece rotation and synchronize the three data streams.

The workpiece is rotated at approximately 10 revolutions per minute (RPM), and approximately 15 – 20 revolutions of data are collected. The data sets for each revolution are averaged to obtain the synchronous portion of the error motions.

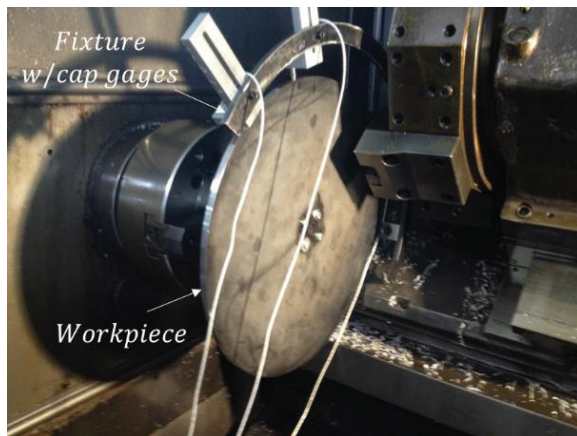


FIGURE 2. Experimental setup for multi-probe testing

RESULTS

Multi-probe Roundness

The results from the multi-probe method are compared to measurements made using a high-precision CMM. Figure 3 shows the comparison.

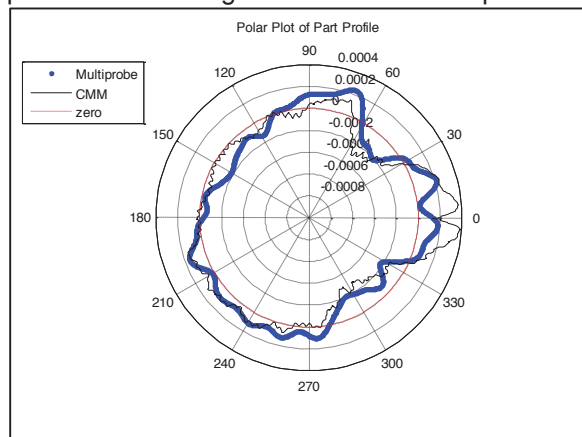


FIGURE 3. Polar plot of Multi-probe and CMM measurements.

The roundness errors obtained using the multi-probe method are created by reconstruction of the workpiece profile from a truncated Fourier series using the first 25 UPR. The CMM data has approximately 1000 points around the circumference of the workpiece. To facilitate comparison, both sets of data are resampled to the same length and filtered using a simple moving average technique to remove high frequencies. The difference between the multi-probe error separation and CMM measurement is shown in Figure 4.

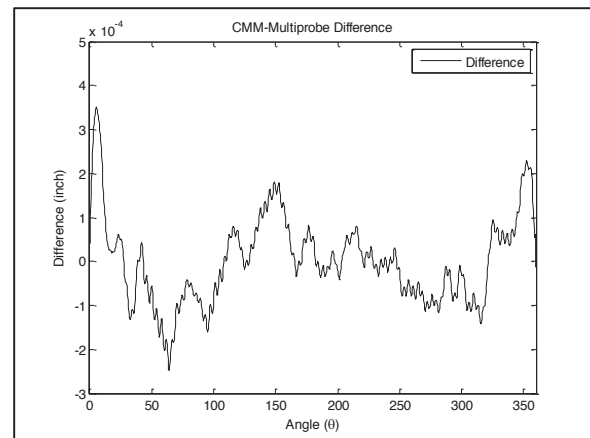


FIGURE 4. Point-to-point difference of Multi-probe and CMM measurements.

The results agree to a max of $+0.0005''$ / $-0.00025''$ ($+0.0127\text{mm}$ / -0.0064mm) band.

The roundness values, calculated as the range of the peak-to-valley of the data, for both the multi-probe error separation and CMM values, which can be seen in Table 1.

TABLE 1. Peak-to-Valley roundness values for multi-probe method and CMM.

Source	Max Value	Min Value	Out-of-Roundness
MP	0.00024"	-0.00023"	0.00047"
	0.0061mm	-0.0058mm	0.012mm
CMM	0.00039"	-0.00032"	0.00071"
	0.0099mm	-0.0081mm	0.018mm

These roundness error values match well and have a difference of $0.00024''$ (0.0061mm).

Uncertainty Budget

The measurand of interest is the roundness error value at discrete angular points around the workpiece. For each error source, the uncertainty for the worse case point is reported in the uncertainty budget. The influence quantities' standard uncertainties, or variations, will be determined either by type A, statistical analysis, or type B, non-statistical analysis, and combined through the method of root-sum-squared to calculate the combined standard uncertainty. A coverage factor of $k = 2$, corresponding to a 95% confidence interval, is used to determine the expanded uncertainty of the measurand under test. This method is the default method of *Guide to the Expression of Uncertainty in Measurement* [6]. The uncertainty sources considered in this work are: 1.) probe angle location, 2.) probe sensitivity variation

including resolution and accuracy 3.) probe plane misalignment, 4.) noise including drift and structural vibrations, 5.) non-normal probe orientation, and 6.) measurement repeatability.

Probe Angle Location

The nominal probe angles were obtained through simulation of the measurement process. The actual probe angles in the as-built device are determined to be 0° , 32.079° and 127.248° . The effect of the probe angle errors on the roundness error measurements is obtained by Monte Carlo simulation using the measured results for spindle error motions and roundness errors from this experiment. Type A is assumed with rectangular distribution, thus a divisor of $\sqrt{3}$ is used. Variations in probe angle location results in a standard uncertainty of 0.003 ($0.0001''$) mm .

Probe Sensitivity, including Resolution and Accuracy

The sensitivities are $467.4732 \frac{V}{in}$, $470.1334 \frac{V}{in}$ and $468.4618 \frac{V}{in}$, which were obtained by measuring a surface, in $0.001''$ ($0.0254mm$) increments, and finding the slope of the curve. The combined standard uncertainty on the calibration report is $12.7 nm$ plus $12.9 \frac{\mu m}{m}$ of range, which includes the resolution error and accuracy. A type B normal distribution is assumed and this value is divided by 1, to get a standard uncertainty of $1.29 \times 10^{-5} m$ ($0.00032''$). It is also noted that these values are for a flat surface calibration and a round workpiece is measured. However, the diameter of the measured workpieces is much larger than the probes' sensing area, so the error is negligible.

Probe Plane Misalignment

Imperfections in the manufacture of the half-ring and brackets holding the probes may cause the probes to not all be aligned in a single plane, and thus be measuring along slightly different circumferential tracks. The error is assumed to be no greater than the total measured roundness error. It is determined that misalignment is about $0.004mm$ ($0.00016''$). The distribution is assumed rectangular, type B, so the value is divided by $\sqrt{3}$. Thus producing a standard uncertainty of $0.0023mm$ ($0.00009''$).

Probe Noise, including Drift and Structural Vibration

The noise in the probes is determined with the probes measuring against a non-rotating workpiece, on the headstock of the lathe with the machine on and the spindle running. Temperature drift and structural vibration are included. The temperature variation in the shop-floor is relatively stable during testing, which takes about 10 minutes or less; so it is assumed to be part of the noise. Each sensor probe measures the non-rotating workpiece to determine the peak-to-average value. The average of the three sensor probe peak-to-average values is $0.0047mm$ ($0.00019''$). A rectangular distribution, type A, is assumed, thus the value is divided by $\sqrt{3}$, giving a standard uncertainty of $0.0027mm$ ($0.00012''$).

Non-normal Probe Orientation

The sensor probe misalignment to the surface normal of the workpiece, can affect the result significantly if the angle of misalignment is significant. In these measurements, the sensor probes were visually aligned to be as normal as possible to the surface to get the best signal. Errors in this alignment create a cosine error. The misalignment error is assumed to be less than $\pm 1^\circ$ with a type B, U-shaped distribution, since a plus-minus variation is used, so the divisor is $\sqrt{2}$. This variation leads to a value of $0.0013mm$ ($0.000052''$). This results in a standard uncertainty of $0.00092mm$ ($0.000036''$).

Measurement Repeatability

The repeatability of the roundness measurement is determined by making multiple measurements. The workpiece is measured once, then the fixture is removed from the turret, then placed back on the turret and the workpiece is measured again. This done two more times to get a total of four roundness measurements. From the four measurements, the spread of the point-to-point standard deviation between the four measurements is used as the value in the repeatability. The value of the repeatability test is $0.00093mm$ ($0.000036''$). The distribution, type A, is assumed rectangular, with a divisor of $\sqrt{3}$. So the standard uncertainty is $0.00054mm$ ($0.00002''$).

The uncertainty contributors are combined in quadrature and multiplied by a coverage factor of 2, for a 95% confidence interval. The uncertainty budget for the roundness measurement can be seen in Table 2.

TABLE 2. *Uncertainty budget for roundness measurement using the Multi-probe error separation method*

Source of Uncertainty	Values (mm)	Type	Probability Distribution	Factor	Standard Uncertainty (mm)
Probe Angle Location	0.0030	A	Rectangular	0.557	0.0017
Probe Sensitivity (Inc. Resolution & Accuracy)	0.0000	B	Normal	1	0.0000
Probe Plane Misalignment	0.0040	B	Rectangular	0.577	0.0023
Noise (Inc. Drift & Structural Vibration)	0.0047	A	Rectangular	0.577	0.0027
Probe Non-normal to Surface of Workpiece	0.0013	B	U-Shaped	0.707	0.0009
Repeatability of Measurement	0.0009	A	Rectangular	0.577	0.0005
Combined Standard Uncertainty (RSS) : $uc(y)$					0.0041
Expanded Uncertainty (k = 2, 95% Confidence) : $U(y)$					0.0082

From the uncertainty budget, it can be seen that there is a large uncertainty in the measurement. The expanded uncertainty, with a 95% confidence is 0.0082mm ($0.0003''$). The dominant uncertainty sources are the noise and the probe non-normal to the surface.

CONCLUSION AND FUTURE WORK

The multi-probe error separation results agree quite well with the CMM. Subsequent evaluation of the machine spindle using a Lion Precision spindle error analyzer system showed relatively large asynchronous errors, 0.043mm ($0.0017''$), in the radial direction for this old and heavily used machine, which may account for some of the differences seen. Nevertheless, the results show that this method may be considered for in-process roundness measurement during machining for large diameter workpieces in shop-floor environments.

The uncertainty budget enables one to understand the primary sources of uncertainty in the measurement. This information is currently being used to help guide the development of new instrument that can measure roundness error of a workpiece, using the multi-probe error

separation method, simultaneously with measuring the workpiece diameter.

REFERENCES

- [1] W.T.Estler, C.J. Evans and L.Z. Shao, "Uncertainty estimation for multiposition form error metrology", *Precision Engineering*, vol. 21, pp. 72-82, 1997.
- [2] M. Coz and A. Lazzari, "Modelling and uncertainty for high-accuracy roundness measurement", in *10th IMKEO TC7 International Symposium*, Saint-Petersburg, Russia.
- [3] S. Cappa, D. Reynaerts and F. Al-Bender, "A sub-nanometre spindle error motion separation technique," *Precision Engineering*, vol. 38, no. 3, pp. 458-471, 2014.
- [4] E. Marsh, B. Knapp and D. Arneson, "Uncertainty sources in Multi-probe error separation.," in *ASPE*, Boston, 2014.
- [5] E. Marsh, *Precision Spindle Metrology*, DEStech Publications, 2010.
- [6] A.Z540.2-1997, *Guide to the Expression of Uncertainty in Measurement*, 1997
- [7] A.B89.3.4M, *Axes of Rotation: Method for Specifying and Testing*, ANSI/ASME, 2010

FORCE ESTIMATION METHOD OF PIEZOELECTRIC ACTUATOR BY USING DRIVING CURRENT

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INTRODUCTION

Because piezoelectric actuators are small and have high frequency response, they are frequently used in small positioning devices. Clamp-and-release type movers such as inchworm-like mechanism [1] and AZARASHI mechanism [2] often use the piezoelectric actuators in their friction control devices. The clamping force should be regulated to stabilize the motion performance. In the case that displacement or force sensors are embedded in the device, the size of the whole device often is large. Various sensor-less control or estimation methods have been proposed. For the displacement estimation, an induced charge feedback [3], real-time permittivity measurement [4] and insertion of a capacitor or resistor [5] have been proposed. Although dynamic force is measured by piezoelectric effect [6], estimation of the static one is difficult [7].

In this paper, the generated force and preload are estimated by using driving current of the piezoelectric actuator.

EXPERIMENTAL SETUP

Figure 1 illustrates the structure of an experimental setup. A stacked piezoelectric actuator (AE0505D16F made by NEC-Tokin, 17.4 μm and 850 N at 150 V), load cell composed of leaf springs with strain gauges and additional load cell for the calibration of static force were arranged in a line. The upper end of the actuator was glued on a frame and the lower one was pressed with an M3 adjustment screw. The whole device was tightened with a C-shaped clamp.

Figure 2 shows the connection of the setup. A step input was applied from a function generator

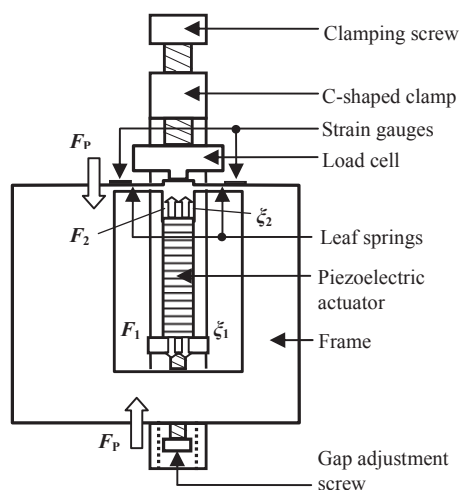


FIGURE 1. Structure of experimental setup.

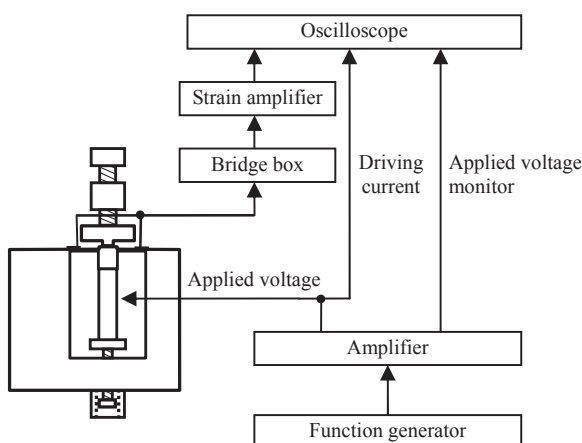


FIGURE 2. Configuration of experimental setup.

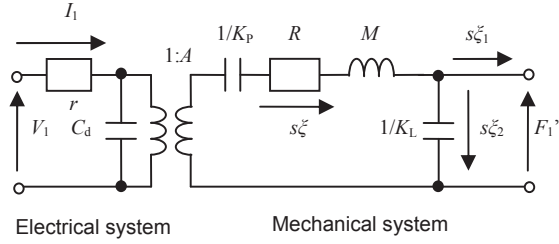


FIGURE 3. Equivalent circuit of experimental setup.

and was magnified with an amplifier ($\times 20$, DC-500 kHz). The force was measured with a 12-bit digital oscilloscope through a strain amplifier (DC-200 kHz). A two-gauge method was used for the temperature compensation. The driving current of the piezoelectric actuator was measured as a voltage drop at a $0.1\text{-}\Omega$ resistor inserted in series.

PRINCIPLE OF GENERATED FORCE ESTIMATION

The force estimation method is composed of two stages: the estimation of dynamic force with an equivalent circuit and preload by resonance frequencies. The following analysis was done as an offline process because the clamping force drifts slowly due to the temperature change.

Estimation of Generated Force Based on Equivalent Circuit

Figure 3 shows an equivalent circuit of a stacked piezoelectric actuator, which is based on Martin's model [8]. The left and right hand sides show the electrical and mechanical systems, respectively. They are connected with a transformer. Table 1 shows the symbols of the parameters used in Fig. 3. The spring constant of the actuator K_p was obtained from the nominal static-characteristics. The capacitance C_d was measured with an impedance analyzer. The others were experimentally obtained from a step response.

The relationship between the driving current and generated force is denoted as:

$$\begin{pmatrix} V_1(s) \\ I_1(s) \end{pmatrix} = \begin{pmatrix} 1 & r \\ 0 & 1 \end{pmatrix} \begin{pmatrix} 1 & 0 \\ sC_d & 1 \end{pmatrix} \begin{pmatrix} \frac{1}{A} & 0 \\ 0 & A \end{pmatrix} \times \begin{pmatrix} 1 & \frac{K_p}{s} \\ 0 & 1 \end{pmatrix} \begin{pmatrix} 1 & R \\ 0 & 1 \end{pmatrix} \begin{pmatrix} 1 & sM \\ 0 & 1 \end{pmatrix} \begin{pmatrix} \frac{1}{K_L} & 0 \\ \frac{K_L}{s} & 1 \end{pmatrix} \begin{pmatrix} F_1'(s) \\ s\xi_1(s) \end{pmatrix}. \quad (1)$$

TABLE 1. Parameters of equivalent circuit.

Symbols	Variables	Values
V_1	Applied voltage	—
I_1	Current	—
ξ_1	Displacement of fixed end ($\xi_1=0$)	—
ξ_2	Displacement of leaf spring	—
F_1'	Dynamic force at fixed end (=Generated force F)	—
A	Force factor	5.75 N/V
C_d	Capacitance of piezoelectric actuator	$1.37 \times 10^{-6} \text{ F}$
K_p	Stiffness of piezoelectric actuator	$4.89 \times 10^7 \text{ N/m}$
K_L	Spring constant of leaf spring	$4.52 \times 10^6 \text{ N/m}$
M	Equivalent mass	$9.00 \times 10^{-2} \text{ kg}$
r	Input resistance of piezoelectric actuator	$0.100 \text{ }\Omega$
R	Damping coefficient	$149 \text{ N}\cdot\text{s/m}$

Then the force and velocity at the fixed end are calculated as:

$$F_1'(s) = \frac{A^2 + C_d(K_p + Rs + Ms^2)}{A} V_1(s) - \frac{A^2 rs + (K_p + Rs + Ms^2)(1 + C_d rs)}{As} I_1(s) \quad (2)$$

$$s\xi_1(s) = \frac{A^2 s + C_d s(K_L + K_p + Rs + Ms^2)}{AK_L} V_1(s) + \frac{A^2 rs + (K_L + K_p + Rs + Ms^2)(1 + C_d rs)}{AK_L} I_1(s) \quad (3)$$

Because the end #1 is open in the equivalent circuit, which means a fixed end, $s\xi_1(s)=0$ in the actual device. Then the following equations are obtained:

$$V_1(s) = \frac{A^2 rs + (K_L + K_p + Rs + Ms^2)(1 + C_d rs)}{A^2 s + C_d s(K_L + K_p + Rs + Ms^2)} I_1(s) \quad (4)$$

$$F_1'(s) = \frac{1}{s} \frac{AK_L}{A^2 + C_d(K_L + K_p) + C_d Rs + C_d Ms^2} I_1(s). \quad (5)$$

Therefore, the generated force F_1' is calculated from the driving current I_1 . However, the preload is not obtained from these equations because the transformer does not transmit the DC component.

Estimation of Preload Based on Resonance Frequency

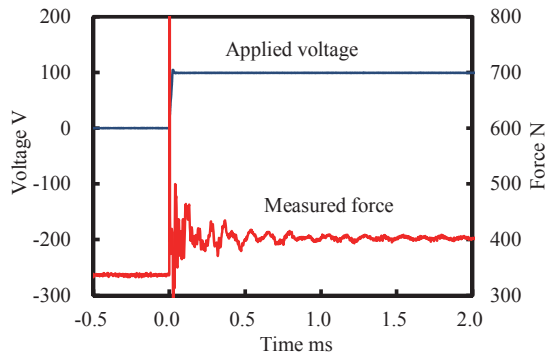


FIGURE 4. Applied voltage and measured force.

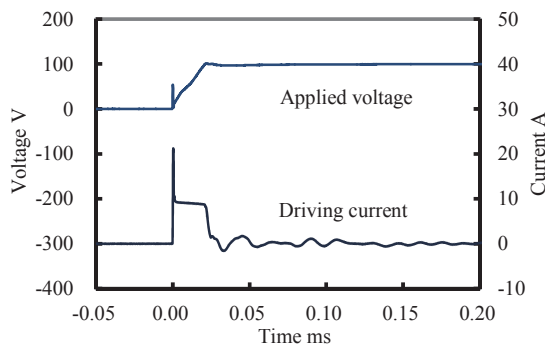


FIGURE 5. Applied voltage and driving current.

The piezoelectric constants d_{31} and d_{33} nonlinearly change against the external force [9, 10]. It has been reported for a bimorph piezoelectric actuator that the resonance frequency was decreased with an increase of the applied voltage [11]. Therefore, it is predicted that the preload affects the spring constant so that the resonance frequency f_R changes accordingly.

EXPERIMENTS

Generated Force vs. Current Waveform and Frequency Components

The generated force and driving current were measured at a preload of 336 N and a step voltage of 100 V. Figure 4 shows the applied voltage and measured force. The residual vibration of the generated force was observed in 2 ms, then the generated force converged at 405 N. Figure 5 shows the expanded view of the applied voltage and driving current. The current was restricted due to the limitation of the

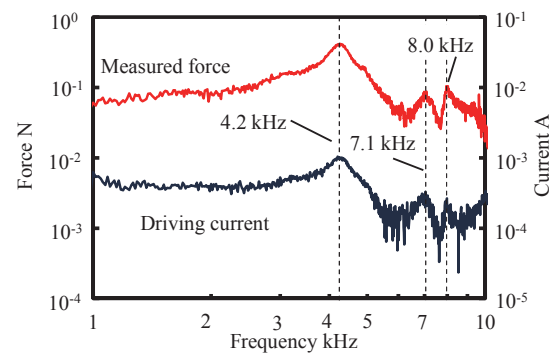


FIGURE 6. Frequency components of generated force and driving current.

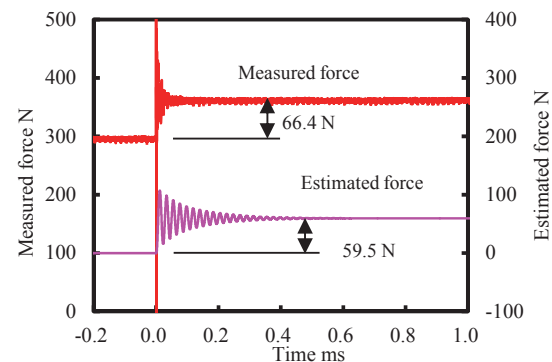


FIGURE 7. Measured and estimated forces.

amplifier in 0.03 ms. Then the current also vibrated.

Figure 6 shows frequency components of the measured force and driving current. The current was analyzed by the fast Fourier transform (FFT) of 34000 points from 0.03 ms with a sampling time of 1 μ s. The frequency resolution was 15 Hz. The first resonance frequency of the generated force, 4.26 kHz was almost the same as that of the current, 4.22 kHz.

Estimation of Force Amplitude by Using Equivalent Circuit

Figure 7 shows measured and estimated forces. The estimated force was calculated by using Eq. (5). The preload and applied voltage were 296 N and 100 V, respectively. Because the equivalent circuit described the first order system, the estimated transient wave form did not coincide with the actual one. The steady-state value of the estimated dynamic force coincided with the actual one. When the preload and applied

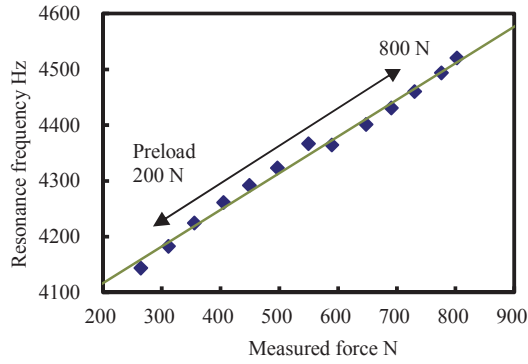


FIGURE 8. Influence of generated force on resonance frequency.

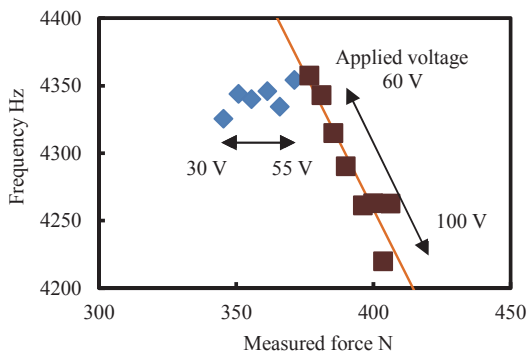


FIGURE 9. Influence of applied voltage on resonance frequency.

voltage were varied as 300, 500 and 700 N, and 60, 80 and 100 V, the maximum error was 9.9%.

Estimation of Force Offset Based on Resonance Frequency

Because the resonance curve fluctuated, it was approximated with a parabola. The driving current follows Eq. (6) around the resonance frequency:

$$I_1 = \frac{k}{\sqrt{(f_R^2 - f^2)^2 + (2Rf_R)^2}}. \quad (6)$$

The resonance frequency was found with the square inverse of Eq. (6):

$$\frac{1}{I_1^2} = \frac{1}{k^2} \left\{ (f_R^2 - f^2)^2 + (2Rf_R)^2 \right\}. \quad (7)$$

Figure 8 shows the relationship between the resonance frequency of the driving current and measured force for various preload. The applied voltage was 100 V. The resonance frequency

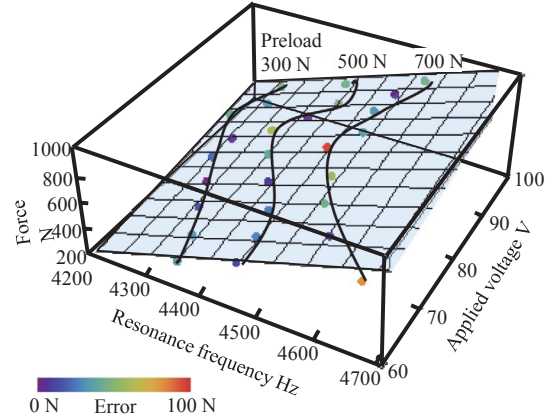


FIGURE 10. Measured force and approximated plane by Eq. (8).

was linearly increased with an increase of the preload from 200 to 800 N.

Figure 9 shows the relationship between the resonance frequency of the driving current and measured force for various applied voltage. The preload was 336 N. The resonance frequency was linearly decreased with an increase of applied voltage from 60 to 100 V. However, the resonance frequency was almost constant below 55 V due to the nonlinearity of the piezoelectric effect [9].

Assuming that the generated force is a function of the applied voltage and resonance frequency, an approximated equation was obtained from the experimental results as follows:

$$F = 1.3159 f_R + 4.3248 V_1 - 5580.4. \quad (8)$$

Figure 10 shows the measured force and approximated plane by Eq. (8). The maximum error was 12.0%. Because the displacement error of the piezoelectric actuator is generally over 10% due to the hysteresis, the estimated error was adequate.

CONCLUSIONS

In this paper, the generated force and preload were estimated by using the driving current of the piezoelectric actuator. Conclusions can be drawn as follows.

- The steady-state value of the dynamic force was estimated by using the equivalent circuit.
- The resonance frequency of the residual vibration of the driving circuit accordingly

changed by changing the preload and applied voltage.

- The generated force was estimated from the applied voltage and resonance frequency of the driving current.

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REFERENCES

- [1] May Jr. WG. Piezoelectric electromechanical translation apparatus. US Patent. 1975; 3902804.
- [2] Furutani K, Hiraoka D, Murase Y, Takeda K. Electrode Feeding Device for Micro-hole Electrical Discharge Machining by Using AZARASHI Mechanism. Proceedings of International Confence of European Society of Precision Engineering and Nanotechnology. 2010; 1: 441-444.
- [3] Furutani K, Urushibata M, Mohri N. Displacement Control of Piezoelectric Element by Feedback of Induced Charge. Nanotechnology. 1998; 9(2) 93-98.
- [4] Kawamata A, Kadota Y, Hosaka H, Morita T. Self-Sensing Piezoelectric Actuator using Permittivity Detection. Ferroelectrics. 2008; 368: 194-201.
- [5] Dosch JJ, Inman DJ, Garcia E. A Self-Sensing Piezoelectric Actuator for Collocated Control. Journal of Intelligent Materials, Systems and Structure. 1992; 3(1): 166-185.
- [6] Badel A, Qiu J, Nakano T. Self-Sensing Force Control of a Piezoelectric Actuator. IEEE Transactions of Ultrasonic, Ferroelectrics and Frequency Control. 2008; 55(12): 2571-2581.
- [7] Takaoka D, Sakaguchi A, Morita Y, Yamada M, Yamaguchi T. Piezoelectric Mount Force of Placement System for Surface Mount Devices. Journal of Japan Society for Precision Engineering. 1997; 63(5): 664-668.
- [8] Martin GE. On the Theory of Segmented Electromechanical Systems. Journal of Acoustic Society of America. 1964; 36(7): 1366-1370.
- [9] Uchino K, Giniewicz JR. Micromechatronics. CRC Press. New York: 2003; 145.
- [10] Takai N, Motogi S, Yamasaki T. Effects of Initial Stresses on Piezoelectric Properties of PZT Disks Analyzed by Nonlinear

Electroelastic Theory. Transactions of Japan Society of Mechanical Engineers A. 2013; 79(807): 1632-1647.

- [11] Wang QM, Zhang Q, Xu B, Liu R, Cross LE. Nonlinear piezoelectric behavior of ceramic bending mode actuators under strong electric fields. Journal of Applied Physics. 1999; 86(6): 3352-3360.

STRAIGHTNESS CONTROL OF A LONG-STROKE, HIGH-PRECISION, SCANNER STAGE

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INTRODUCTION

This paper summarizes a new straightness control algorithm, developed and applied to a long-stroke, high-precision scanner stage.

The stage in question is controlled in 3 axes with laser interferometer feedback; X, Y and Yaw. Bar mirrors installed on the stage are used for interferometers' reflection. The desired stroke of the stage is longer than maximum length of bar mirror currently available commercially. In order to enable use of interferometers with this limitation, multiple interferometers are installed and used for single axis control by switching between the two interferometers in real-time. A switching correction algorithm is designed and implemented on the controller to minimize the effect of dynamic yaw on the switching error.

SYSTEM CONFIGURATION

Figure 1 shows a simple representation of the equipment. Precision stage with 3 axis control is used in the equipment.

Figure 2 is a 2-d representation of stage axes, motors and interferometers. Stage is controlled with interferometers for improved accuracy and repeatability.

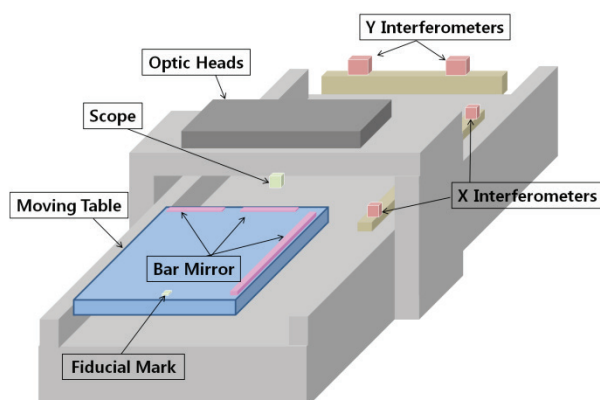


FIGURE 1. Simple Representation of Equipment

The moving table on which substrate is placed scans along Y axis with a full stroke. Stage dynamic yaw is controlled by adjusting current to the Y2 motor. Multiple scans are necessary to

cover the full range of a large size substrate which is achieved by scanning in Y axis at multiple X positions.

Y interferometers in this equipment are operated in differential mode with respect to the optical head block. This way any displacement of optical head block can be compensated by interferometer itself.

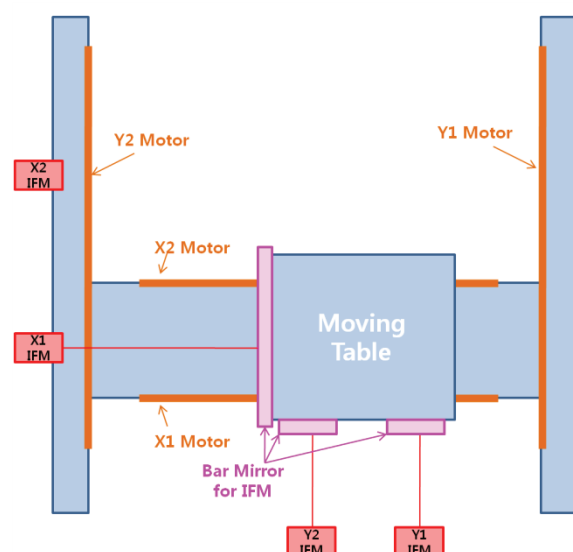


FIGURE 2. Simple Representation of Stage Axes Configuration

Due to the size of substrate and distance to be covered by optical heads and for calibration tasks and loading/unloading of substrate, the desired stroke in Y direction is about 4.8m. To cover this unusually long stroke with laser interferometers, the bar mirror should be of length that is longer than the stage stroke, which is not available with current material or manufacturing processes.

To address this limitation, stage control was designed by employing two laser interferometers in X-axis (X1 IFM and X2 IFM) as shown in Figure 2 which controls the straightness of stage during scanning. At any given time one of two X

interferometers takes control depending on the position of the stage in Y direction.

This way, the bar mirror length can be only half of the stroke, which makes its manufacturing possible and its assembly to the stage more favorable.

CONCEPT OF SWITCHING BETWEEN INTERFEROMETERS

Switching between two interferometers requires a control algorithm which operates fast enough in order to ensure control feedback is correctly switched when stage position crosses-over from one interferometer active region to the other. Simply put, one half of full Y stroke is covered by X1 IFM and other half by X2 IFM. There should however be an overlap region wherein both interferometers are active in order to handover the control of stage from interferometer to the other. Figure 3 illustrates the switching concept.

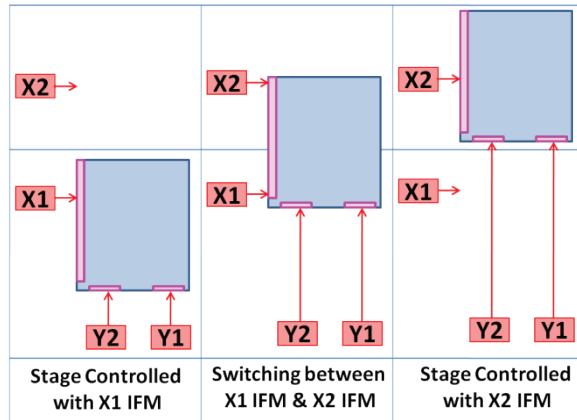


FIGURE 3. Concept of Switching between Multiple Feedback Interferometers

At the instant of switching between one interferometer to other, a switching offset is calculated in controller based on position feedbacks of two interferometers. This offset is added to position feedback, and the resulting position data is used in control loop.

$$P_{CONFB} = P_{XCUR} + P_{OFFSET} \quad (E1)$$

where

P_{CONFB} : Position feedback in control loop

P_{XCUR} : Feedback from currently active X interferometer

P_{OFFSET} : Offset calculated at switching instant according to (E2)

$$P_{OFFSET, NEW} = P_{OFFSET, CUR} + (P_{XNEW} - P_{XOLD}) \quad (E2)$$

where

$P_{OFFSET, NEW}$: New switching offset

$P_{OFFSET, CUR}$: Current switching offset

P_{XNEW} : Feedback from X interferometer that will be used after switching

P_{XOLD} : Feedback from X interferometer currently being used (before switching)

REPEATABILITY ERROR ASSOCIATED WITH SWITCHING

Implementing the concept with just the calculations described in last section results in error every time switching occurs. This switching error is primarily the effect of stage's dynamic yaw error which in our precision stage can be up to $1\mu\text{m}$ depending on the stage's speed of scanning. Figure 4 illustrates the effect of dynamic yaw on switching error. As shown in the figure, depending on stage yaw posture at the instant of switching, an error can result.

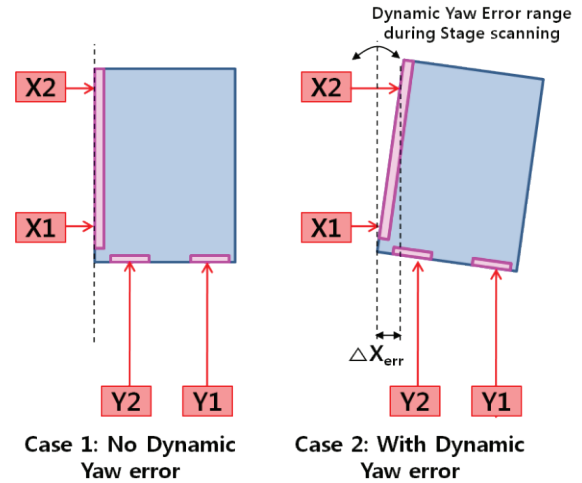


FIGURE 4. Effect of Stage Dynamic Yaw Error on Switching Error

The switching error is typically dependent on stage speed and also the type of motion profile. In our system, maximum switching error of about $0.5\mu\text{m}$ occurred at 60mm/s stage speed while at 300mm/s , the error shot up to over $2\mu\text{m}$ per switching.

In addition, diagonal motion of stage wherein stage moves simultaneously in both X and Y directions results in larger switching error than straight Y motion. Diagonal motion at 60mm/s results in $2.5\mu\text{m}$ error and at 300mm/s , over $12\mu\text{m}$ per switching. Errors of this order are naturally unacceptable in a precision stage where repeatability of less than $0.2\mu\text{m}$ is desired.

DYNAMIC YAW CORRECTION BASED SWITCHING ALGORITHM

An algorithm to reduce the effect of dynamic yaw on switching error was designed and successfully implemented on the stage. Application of this algorithm reduced the switching error to less than $0.2\mu\text{m}$ down from $2\mu\text{m}$ even at maximum stage speed of 300mm/s . The algorithm takes into account the dynamic yaw of stage while calculating the feedback position used in the control loop. In an ideal system with no mechanical deformation, the correction factor due to yaw would be a linear equation of real time yaw. However, in a real system with mechanical deformations are large enough to require higher order calculations. This results in another problem since the interferometer feedback calculations in question are performed in FPGA with a high speed clock, and the more complex the calculations, the larger latency it causes in position feedback to the controller. Thus an optimization is required to ensure both constraints are met. A simplified version of yaw correction based switching algorithm would look something like equation below.

$$P_{CONFB} = P_{XCUR} + P_{OFFSET} + f(\text{yaw}, y, x) + M_{IFM} \quad (E3)$$

where

P_{CONFB} : Position feedback in control loop

P_{XCUR} : Feedback from currently active X interferometer

P_{OFFSET} : Offset calculated at switching instant according to (E2)

$f(\text{yaw}, y, x)$: Correction factor due to real-time yaw

M_{IFM} : Correction factor due to mechanical deformations and displacements

The $f(\text{yaw}, y, x)$ in equation (E3) consists of two components. First component is the first order correction with respect to stage yaw. Second is component representing the deformation of the bar mirror during stage motion. The latter can be negligible depending on the rigidity of the mechanical parts of stage particularly the moving table and bar mirror.

M_{IFM} represents the effect of the deformations and/or displacements of interferometer blocks and that of optic head block which should also be compensated in order to minimize the switching error.

This factor can be measured using another set of interferometers, e.g., IFM_{Aux_X1} , IFM_{Aux_X2} ,

IFM_{Aux_Y1} , and IFM_{Aux_Y2} . These interferometers measure the relative deformations and displacements of the optic head block with respect to interferometers' blocks on which X and Y control interferometers are installed.

The reason this factor is required is because the yaw used for switching correction algorithm is calculated as difference of two Y interferometers while those Y interferometers are operated in a differential mode (i.e., they measure the relative motion of stage with respect to the optic head block). As stage moves in scan direction, there might be deformation and/or displacement of optic head block which, if not measured and compensated, can cause erroneous correction.

The result of switching error with and without correction algorithm is shown in Figure 5. For this measurement, stage is moved back and forth along scanning axis (Y), and repeatability of stage is measured along switching axis (X) by measuring position of a fiducial mark using a scope (see Figure 1) after each switching. Each index of X axis in the plot represents one back and forth motion which results in two times of switching; once during forward scan motion and once during reverse scan motion.

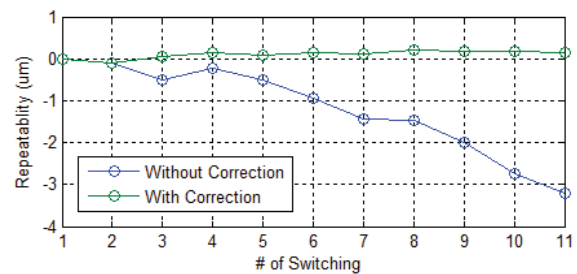


FIGURE 5. Result of Switching Error Measurement Test

As shown in the Figure 5, not applying switching correction can result up to about $0.9\mu\text{m}$ error for a back and forth motion of stage in scan direction. This error gets accumulated for each scan motion, and for a 10 scan motion sequence as in our system, it can result up to $4\mu\text{m}$ repeatability error. With application of switching error, this is reduced to under $0.2\mu\text{m}$ for 10 scan motion as shown in Figure 5.

CONCLUSION

Precision stages typically employ interferometer feedback control. However, for a large size stage with long stroke, a bar mirror to cover full stroke may not be commercially available as in the case of our system. For this reason, multiple interferometers are used for a single axis control

and switching between two interferometers in real time depending of the stage position. This switching results in an error which is caused by dynamic yaw of the stage as well as non-linear mechanical deformation and/or displacement. The axis in question controls the straightness of stage during scanning, so switching error results in non-repeatable straightness profile of stage for each scan. A correction algorithm was developed and implemented to reduce the switching error and hence the straightness profile error. Applying this algorithm reduced the error from about $4\mu\text{m}$ to less than $0.2\mu\text{m}$ which was below the allowed specification for repeatability error of the stage.

REFERENCES

- [1] J. Kim, S. Park, T. Son, S. Jang, Real-time straightness motion error measurement and compensation system for the 8G high-resolution flat panel manufacturing machine, American Society for Precision Engineering, 2014.
- [2] Laser and Optics User Manual, Agilent.

ON-LINE TEMPERATURE PREDICTION OF A PERMANENT MAGNET TUBULAR LINEAR MOTOR FOR CONTROL PARAMETER TUNING

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INTRODUCTION

Linear motors have been employed in various industrial applications to provide linear motion directly. To ensure high control performance, some control algorithms (e.g. model reference control, observer) need to know exact motor parameters for control parameter tuning. Some constant parameters such as inductance and load mass can be measured through off-line methods. However, for temperature dependent parameters, such as coil resistance and magnet remanence, they should be measured in real-time when the linear motor operates in a wide range of temperature. This paper aims to propose an on-line temperature prediction approach to predict the temperature rises in different parts of a permanent magnet tubular linear motor (PMTLM) with no use of extra temperature sensors. Consequently, the temperature dependent parameters can be estimated and accordingly the control parameters can be tuned in real-time.

Various numerical and analytical approaches can be used for the purpose of temperature prediction [1-3]. Although numerical ones such as finite element method are better at dealing with complex cases, they are time-consuming and inconvenient for optimization. For this reason, the analytical approach lumped-parameter thermal modeling is adopted in this paper to ensure high computation efficiency in real-time. In [4], a one-dimensional thermal network model is adopted, where heat flows radially to the ambient only from the external surface of the stator. Ref. [5] considers also heat exchange on parts of the translator that are not surrounded by the armature. Moreover, a detailed thermal network model given in [6] involves the axial temperature variation in the armature. The solution accuracy depends on the complexity of the lumped-parameter model. The thermal network model established in this paper differs from previous ones due to some variations in configuration and installation.

The rest of this paper is organized as follows: Section 2 describes the basic structure of the PMTLM under study. Its thermal network model is presented in Section 3. An iterative computation process which combines the control model and the thermal network model is put forward in Section 4. The experiment for validation is given in Section 5 and this paper is concluded in Section 6.

PERMANENT MAGNET TUBULAR LINEAR MOTOR

The PMTLM under study is shown in Figure 1 and Figure 2. The fixed part (stator) consists of a polymer tube, windings and a back iron from the inside out. The moving part (translator) is composed of axially magnetized permanent magnets and iron pieces. The main parameters of the PMTLM are given in Table 1.

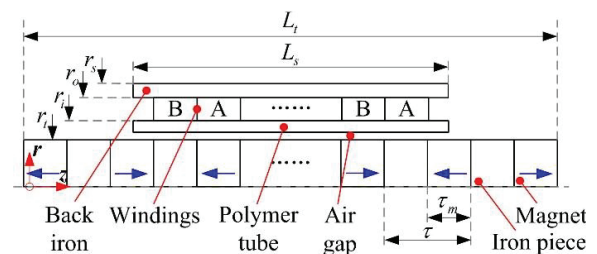


FIGURE 1. Basic structure of the PMTLM

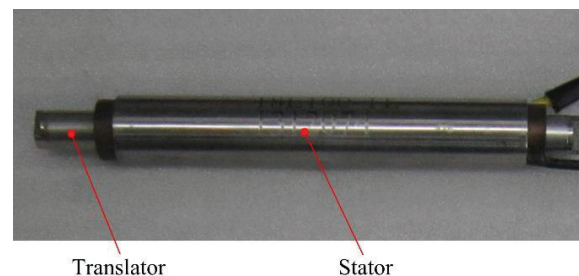


FIGURE 2. The PMTLM prototype

TABLE 1. Parameters of the PMTLM prototype

Sym bol	Description	Value (mm)
L_s	Length of the stator	105.0
L_t	Length of the translator	140.0
r_s	Outer radius of the stator	9.0
r_o	Outer radius of the windings	7.8
r_i	Inner radius of the windings	5.6
r_t	Outer radius of the translator	5.0
τ_m	Length of the magnet	8.0
τ	Pole pitch	10.0

LUMPED-PARAMETER THERMAL NETWORK MODEL

The lumped-parameter thermal network model is similar to an electric circuit: temperature to voltage, power to current, thermal resistance to electric resistance and thermal capacitance to electric capacitance [1]. First, the motor structure should be divided into several segments connected through nodes. Each segment is represented by a thermal resistance and a thermal capacitance. Due to the axially symmetric configuration, the motor structure can be viewed as a combination of a solid cylinder (i.e. translator) and a series of hollow cylinders (i.e. air gap, polymer tube, windings and back iron) from the inside out. Therefore, by integrating the models of some basic geometric shapes, the lumped-parameter thermal network model of the PMTLM is established, as shown in Figure 3. Both radial and axial heat transfers are considered. Only half is modeled due to the symmetry.

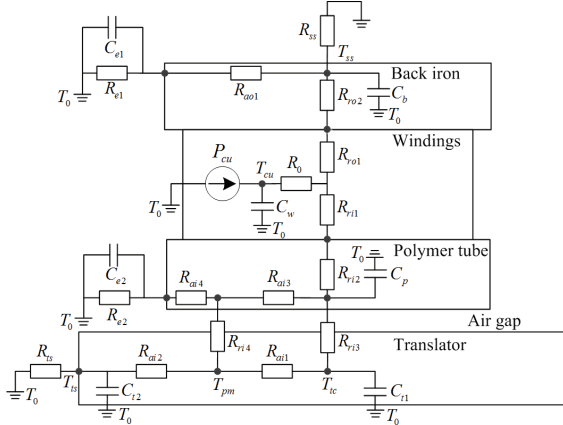


FIGURE 3. Lumped-parameter thermal network model of the PMTLM

The main work to solve the thermal network model is the determination of the thermal

capacitances and the thermal resistances associated with conduction, convection and radiation. A conduction thermal resistance relies on the knowledge of the geometric shape and the material thermal conductivity. A convection thermal resistance depends on the corresponding convection coefficients. The radiation is neglected due to the low emissivity of the surfaces. The formulas for calculating these parameters are given in Table 2.

TABLE 2. Parameters of the thermal network model of the PMTLM

Symbol	Formula
R_{ro1}	$\frac{1}{4\pi\lambda L_a} \left(1 - \frac{2r_i^2 \ln\left(\frac{r_o}{r_i}\right)}{(r_o^2 - r_i^2)} \right)$
R_{ri1}	$\frac{1}{4\pi\lambda L_a} \left(\frac{2r_o^2 \ln\left(\frac{r_o}{r_i}\right)}{(r_o^2 - r_i^2)} \right)$
R_0	$\frac{-1}{8\pi(r_o^2 - r_i^2)\lambda L_a} \left(r_o^2 + r_i^2 - \frac{4r_o^2 r_i^2 \ln\left(\frac{r_o}{r_i}\right)}{(r_o^2 - r_i^2)} \right)$
R_{ro2} R_{ri2} R_{ro3} R_{ri4}	$\frac{1}{2\pi\lambda L_a} \ln\left(\frac{r_o}{r_i}\right)$
R_{ao1} R_{ai1} R_{ai2} R_{ai3}	$\frac{L_a}{2\pi\lambda(r_o^2 - r_i^2)}$
R_{fc} R_{nc}	$\frac{1}{hA}$
R_{e1} R_{e2}	Determined experimentally
$C?$	$c_p m$

Once all the thermal capacitances and resistances are obtained, each node temperature indicated in Figure 3 with respect to time can be calculated based on electric circuit principles:

$$\begin{bmatrix} T_{cu}^{(i)} \\ T_{ss}^{(i)} \\ T_{ic}^{(i)} \\ T_{pm}^{(i)} \\ T_{ts}^{(i)} \end{bmatrix} = \begin{bmatrix} T_{cu}^{(i-1)} \\ T_{ss}^{(i-1)} \\ T_{ic}^{(i-1)} \\ T_{pm}^{(i-1)} \\ T_{ts}^{(i-1)} \end{bmatrix} + \begin{bmatrix} \Delta T_{cu} (P_{cu}^{(i)}) \\ \Delta T_{ss} (P_{cu}^{(i)}) \\ \Delta T_{ic} (P_{cu}^{(i)}) \\ \Delta T_{pm} (P_{cu}^{(i)}) \\ \Delta T_{ts} (P_{cu}^{(i)}) \end{bmatrix} \quad (1)$$

where T_{ic} , T_{ts} and T_{ss} represent the temperatures of the translator center, the translator surface and

the stator surface respectively, T_{cu} is the mean temperature of the copper, and T_{pm} is the mean temperature of the permanent magnets. The heat power P_{cu} resulting from the copper loss is injected into the thermal network model at the mean temperature node of the windings. It satisfies

$$P_{cu}(t) = i^2(t) R(t) \quad (2)$$

As stated above, the resistance R and the magnet remanence B_r are both temperature dependent as follows:

$$R(t) = \bar{R} (1 + \alpha_{cu} (T_{cu}(t) - T_0)) \quad (3)$$

$$B_r(t) = \bar{B}_r (1 + \alpha_{pm} (T_{pm}(t) - T_0)) \quad (4)$$

where α_{cu} and α_{pm} are the temperature coefficients of the copper and the permanent magnets respectively.

As the force constant K is in direct proportion to the magnet remanence, one has

$$K(t) = \bar{K} (1 + \alpha_{pm} (T_{pm}(t) - T_0)) \quad (5)$$

Thus, all the temperature dependent parameters can be estimated in real-time.

COMBINATION OF THE CONTROL MODEL AND THE THERMAL NETWORK MODEL

Figure 4 gives an example of the combination between the control model (Figure 4(a)) and the thermal network model (Figure 4(b)). An iterative computation process is put forward as follows. The heat loss P_{cu} injected into the thermal model

is determined by multiplying the coil resistance $R(t)$ by the current squared $i^2(t)$. The adjustment mechanism in the control model updates the control parameters according to the coil resistance $R(t)$ and the force constant $K(t)$, which depend on the mean temperatures of the windings and the magnets respectively.

EXPERIMENT

The experimental setup for validation is shown in Figure 5. A load supported by a linear guide is attached to the end of the PMTLM. Two platinum-based resistance temperature sensors (PT100, resolution 0.5 °C) are mounted on the stator and translator surfaces to measure T_{ss} and T_{ts} respectively. The temperature monitor is to display and record the value of each temperature sensor. As the radial thermal resistances are much smaller compared with the axial ones, T_{ss} , T_{tc} , and T_{cu} are very close to each other. So T_{ss} can be used to estimate T_{tc} and T_{cu} such that no extra sensors are needed to place inside the PMTLM. T_{pm} can be estimated by averaging T_{tc} and T_{ts} . The ambient temperature T_0 is about 23 °C.

In the experiment, a constant current of 0.5 A is fed into the windings. The values of the two temperature sensors are recorded every minute. Figure 6 displays the comparison between the predicted and measured temperatures; it is seen that they two match well with the maximum error of 2 °C.

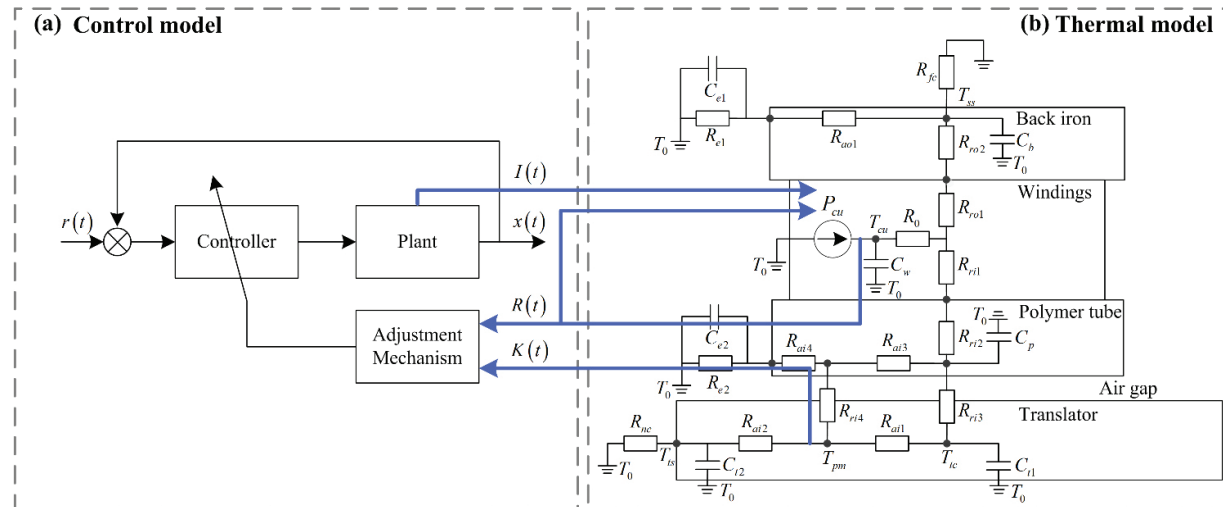


FIGURE 4. On-line temperature prediction for control parameter tuning based on an iterative computation process

CONCLUSIONS

This paper proposes a lumped-parameter thermal network model of a permanent magnet tubular linear motor (PMTLM), which provides a rapid and accurate approach for on-line temperature prediction. The prediction error stays below 2 °C. Consequently, the temperature dependent parameters of the PMTLM such as coil resistance and magnet remanence can be estimated and accordingly the control parameters can be tuned in real-time.

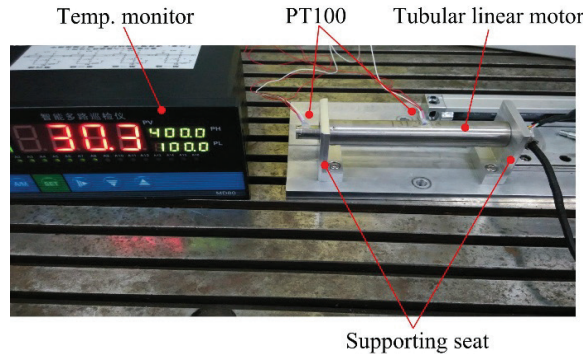


FIGURE 5. Experimental setup

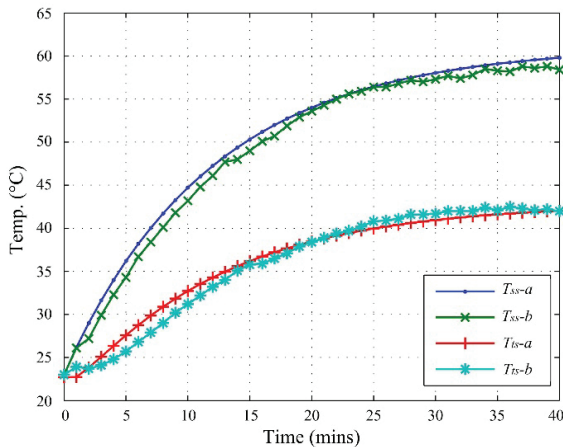


FIGURE 6. Comparison between predicted and measured temperature rises (a: Predicted; b: Measured)

ACKNOWLEDGEMENTS

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REFERENCES

- [1] A. Boglietti, A. Cavagnino, D. Staton, et al, "Evolution and modern approaches for thermal analysis of electrical machines," *IEEE Transactions on Industrial Electronics*, vol. 56, no. 3, pp. 871-882, 2009.
- [2] L. Alberti, N. Bianchi, "A coupled thermal-electromagnetic analysis for a rapid and accurate prediction of IM performance," *IEEE Transactions on Industrial Electronics*, vol. 55, no. 10, pp. 3575-3582, 2008.
- [3] X. Huang, J. Liu, and C. Zhang, "Calculation and experimental study on temperature rise of a high overload tubular permanent magnet linear motor," *IEEE Transactions on Plasma Science*, vol. 41, no. 5, pp. 1182-1187, 2013.
- [4] Vese, I.-C., Marignetti, F., Radulescu, M.M., 2010, "Multiphysics approach to numerical modeling of a permanent-magnet tubular linear motor", *IEEE Transactions on Industrial Electronics*, vol. 57, pp. 320-326.
- [5] Ridge, A.N., Mathekga, M.E., Clifton, P.C.J., et al, 2012, "Thermal modelling of a tubular linear machine for marine renewable generation", 6th IET International Conference on Power Electronics, Machines and Drives (PEMD 2012), pp. B144.
- [6] Ridge, A., McMahon, R., and Kelly, H.-P., 2013, "Detailed thermal modelling of a tubular linear machine for marine renewable generation", *IEEE International Conference on Industrial Technology (ICIT)*, pp. 1886-1891.

DYNAMIC MODELING OF AN H-TYPE PRECISION GANTRY INCLUDING BASE MOTION EFFECTS

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INTRODUCTION

This manuscript presents preliminary work on a multi-axis frequency-domain model of a precision H-bridge gantry system, and is intended to demonstrate the use of a data-based methodology [1]. Gantry-style designs are extremely common in industrial automation systems, and are often used for electronics manufacturing, laser machining, and additive-manufacturing processes. Figure 1 shows the Aerotech AGS15000 gantry system used for the experimental portion of this work. The system has travels of 600 mm in the

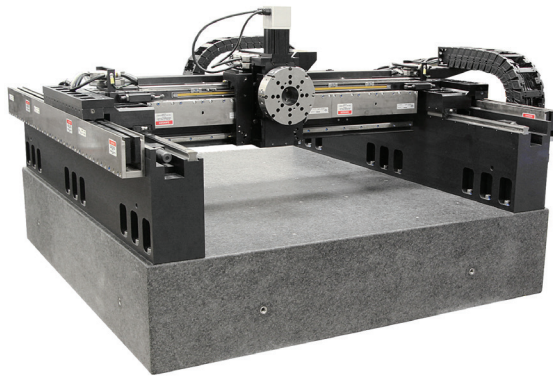


FIGURE 1. This Aerotech AGS15000 is typical of the configuration for a precision gantry system. bridge (X) axis, and 1000 mm in the parallel gantry (Y,YY) axes. This stage system also includes a Z-axis and a simulated customer payload, but they were not actively used in the testing. Applications requiring high accuracy and fast response times almost always include dual linear-motor actuation and feedback on the gantry axis, which creates a coupled servo control problem which becomes the main focus of this work. The gantry is idealized as a planar system consisting of a single bridge axis (X) along with parallel gantry axes (Y and YY).

Gantry systems are familiar-enough to make them a useful example of a multi-input, multi-output control program. This manuscript follows most closely the techniques of Gordon and

Erkorkmaz [2] for analyzing and interpreting the measured responses. Other relevant past works include García-Herreros [3] which develops a model of an H-type gantry that includes torsional compliance at both ends of the gantry bridge, and Teo [4] which incorporates the moving slider mass into an inertia-based model.

EXPERIMENTAL PROCEDURE

Frequency response techniques are used extensively at Aerotech to characterize the behavior of motion systems. Tools for measuring the frequency response of single-input, single-output servo systems are built into our standard controller software, but capturing a fully-multivariable response required developing some additional tools. The following technique provided the experimental data presented in this manuscript.

1. Move the axes under control to a predefined location in the gantry workspace.
2. Disable the servo control for all axes.
3. Command a sinusoidally-varying current at a predefined frequency to a single motor.
4. Record the measured positions from all three encoders, and the accelerations from an arrangement of three base-mounted inertial sensors.
5. Re-enable the servo control, and return all axes to the position setpoint. This lessens the effect of accumulated position drift.
6. Repeat the process for each of three motors over a range of frequencies and at different locations.

The input frequency ranged from 5 Hz to 1000 Hz at 200 logarithmically-spaced steps, and the current amplitude was 1.5 A, corresponding to a force of about 60 N. Each X-axis excitation was performed at five different Y-axis locations, and vice-versa, generating a total of three thousand individual data files. The entire process is scripted to

run largely unattended and required about three hours to complete. MATLAB is then used to post-process the data files, extracting the magnitude and phase of the measured motion in response to the commanded current at that frequency - thus creating a single point in the multivariable response. The MATLAB `frd` command is extremely useful for combining all of the experimental results into a single data structure for further analysis.

This data collection was somewhat excessive and could be minimized for routine production use. Real-time processing of the data sets would eliminate the need to save files of each time trace, White-noise inputs would also be much faster, but our experience is to prefer sinusoidal inputs for testing most mechanical systems. A more-complete set of experiments would have had measurements taken over a full XY grid (or even XYZ volume), and also over a range of drive levels to capture the nonlinear behavior of the frictional force from rolling-element bearings [5].

MEASURED RESPONSES

The as-measured responses create a 3×3 system matrix which converts motor forces to displacements at the linear encoders. Figure 2 shows preliminary frequency response data of the displacement when driven by the force from each particular motor, and the main characteristics are similar to those reported by others in the literature. The X-axis (i.e. the bridge axis) has a response similar to a free mass up until what appear to be three resonances in the neighborhood of 200 Hz. We also see the expected symmetry in the responses at the Y and YY encoders when driven by each of the Y and YY motors. They nicely demonstrate the classic collocated versus non-collocated vibration example. The displacement at each encoder, when driven by the motor on the same side, shows a pair of complex-conjugate zeros preceding a pair of complex-conjugate poles. This puts an anti-resonance before the main resonant frequency. When driven and measured on opposite sides, the response only has the pair of poles. These are at the main “yaw-mode” frequency where the two sides of the gantry bridge move in opposite directions.

It is interesting to note that the axis location had very little influence on the main responses. That observation should not be generalized, but was a consequence here of a fairly light carriage ($m_x =$

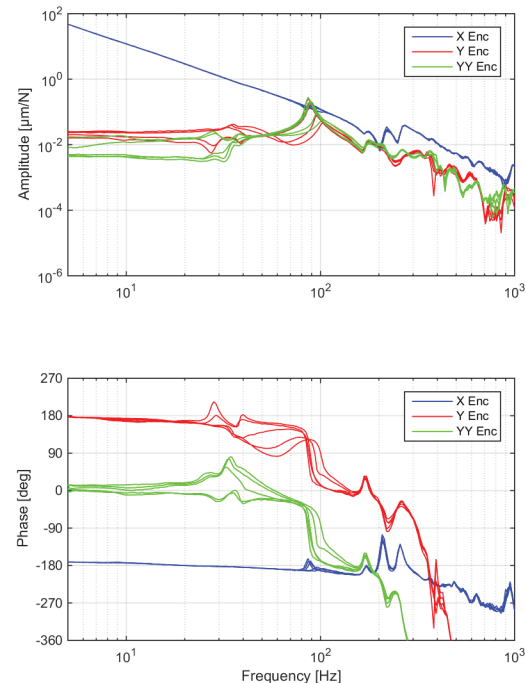


FIGURE 3. Frequency response measurements show generally the same magnitude of motion at all encoders when driven by the X-axis motor.

21 kg) moving on a bridge with the densest elements (the copper motor coils) located at the farthest distance from the center of mass. The X-axis is often considered to be decoupled from the Y and YY axes. This is because it moves orthogonally to them, and does not have an obvious mechanism for coupling its motor forces into Y-direction motion.

This gantry did show a strong coupling between X-axis motor forces, and Y and YY displacement measurements. The point where the X-axis motor force is applied is offset from the center of mass of the gantry axis, and so a force applied by the X-axis motor creates a rotation of gantry bridge. The coupling is strong enough that in the neighborhood of the main yaw mode, more motion is generated at the Y and YY encoders than at the X-axis encoder by an X-axis force. Figure 3 shows the responses at the X, Y, and YY encoders when the system is driven by an X-axis force. This implies that “Yaw Mode” or “R-Theta” control techniques that act only on the gantry axes do not correctly decouple all of the axis behavior.

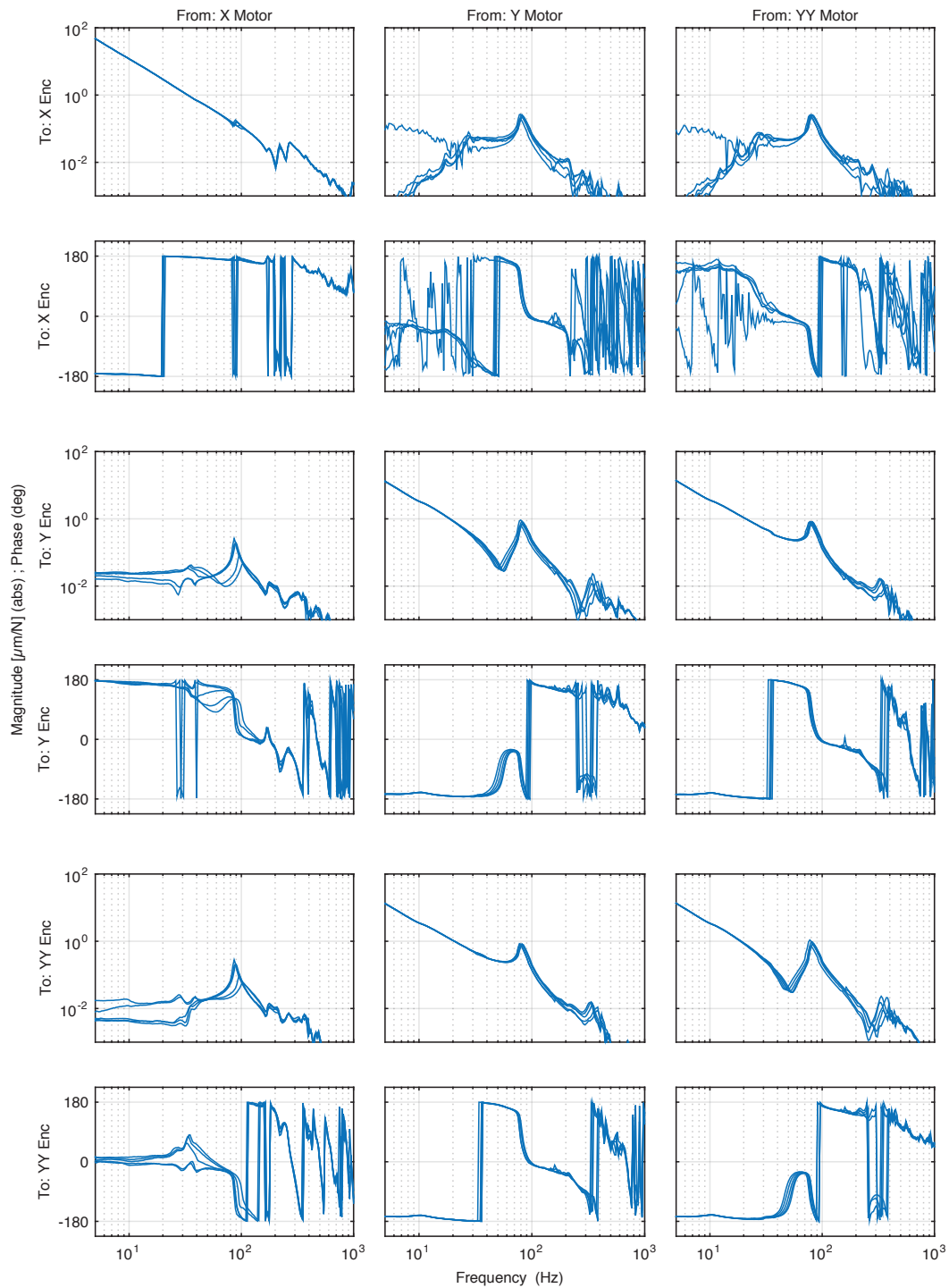


FIGURE 2. Preliminary experimental frequency response measurements (displacement per force) from the AGS15000 show the expected symmetry in the gantry axes, Y and YY, as the vibration modes are either collocated (measured near to the point where the force is applied) or non-collocated (measured on the opposite side). The responses also show spring-like dynamics at low frequencies in the carriage, X, as oscillations of an offset payload couple into yaw of the bridge. The overlapping traces represent measurements taken at different locations in the stage travel.

AXIS DECOUPLING

The highly-coupled nature of the Y and YY axes is often addressed through the use of a coordinate transform to decouple them, and effectively diagonalize the transfer function matrix. Adapting the presentation of Gordon and Erkorkmaz [2], we will define a kinematic transformation between encoder measurements and displacements at the geometric center of the gantry bridge as

$$\begin{bmatrix} y_1 \\ y_2 \\ y_3 \end{bmatrix} = \mathbf{T} \begin{bmatrix} \bar{x} \\ \bar{y} \\ \theta \end{bmatrix}, \quad \mathbf{T} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & a_1 \\ 0 & 1 & -a_1 \end{bmatrix}. \quad (1)$$

Measurements taken at the encoder positions are given by y_1 , y_2 , and y_3 , and the distance a_1 is the distance from the center of the gantry to the Y and YY encoders. Likewise, we convert the forces applied by individual motors to equivalent body-centered forces through the transformation,

$$\begin{bmatrix} u_x \\ u_y \\ u_\theta \end{bmatrix} = \mathbf{F}_q \begin{bmatrix} f_1 \\ f_2 \\ f_3 \end{bmatrix}, \quad \mathbf{F}_q = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 1 \\ a_2 & a_3 & -a_4 \end{bmatrix}. \quad (2)$$

The parameters a_2 , a_3 , and a_4 are the distances between the center of mass of the Y-axis and the point where the forces is applied. In a strictly-correct transformation, the bridge yaw angle is now expressed in radians (or microradians) and the applied torque expressed in units of newton-meters. However, it often seems easier to understand in practice to express the units as micrometers of offset between the two parallel gantry encoders per differential force applied by the motors. This is accomplished with a coordinate change.

Applying these transformations to the measured frequency response data generates an updated transfer function matrix from the virtual (combined/effective) body forces to coordinates at the geometric center of the bridge. Figure 4 show the new three-by-three set of frequency response plots. The X-axis response is largely unchanged, but what were the Y and YY responses are now much more decoupled. The Y-axis response now clearly shows a second co-located resonant mode at about 345 Hz, which is likely a pitching motion of the Z-axis stage and payload about the gantry bridge. The Theta response is the most substantially-different, now appearing as a flat spring-like response below the approximately 80 Hz yaw-mode resonance, and attenuating as a mass-like system after that. The off-axis responses are lower, and the phase especially is

becoming more irregular. This apparent randomness indicates that we need to increase the model complexity - likely through quantifying the effect of the shifting center of mass over travel - in order to further increase the level of decoupling.

MACHINE BASE MOTION

The base-mounted accelerometers show how the granite and machine frame move in response to the applied motor forces. As a reminder, all motor forces applied to a motion system apply equally and in the opposite direction to the machine base. Linear encoder measurements are therefore always a relative measurement between a moving stage and a moving base. Figure 5 shows an overlay of just the main on-axis responses. The measured base acceleration is numerically integrated into displacement for an easier comparison with the encoder measurements. The isolation system for this granite base is a set of elastomer pads between the granite and a steel weldment. The plots show their resonant frequency to be at a somewhat well-damped 10 Hz frequency. Several other items to note are the offsets in the responses (as predicted by the ratio of the moving masses), the prominent resonance in the X-direction response at 200 Hz (likely due to a lateral motion of the gantry bridge in the X-direction, and the coupling in Theta at the main yaw-mode resonance. This system had a very substantial granite base weighing approximately 1600 kg, and the coupling would be more acute if the design dictated a thinner or lighter base. These sorts of measurements can be used to quickly guide the decisions on whether the designer faces a "stage problem" or a "base problem."

CONCLUSION

A detailed picture of the behavior of a gantry system is possible with a limited set of multivariable frequency response plots. From this point, one could follow a data-driven methodology to apply control algorithms to each of the substantially-decoupled axes via conventional loop shaping techniques. A lumped-mass model could also be developed to further guide design decisions and to carefully account for the differences between inertial and relative displacements. Finally, these transformations show the response at the geometric center of the bridge. More interesting to many users is the location of a moving point, offset from the bridge and carriage location (i.e. where their end-effector is located) which would require a position-based coordinate transformation.

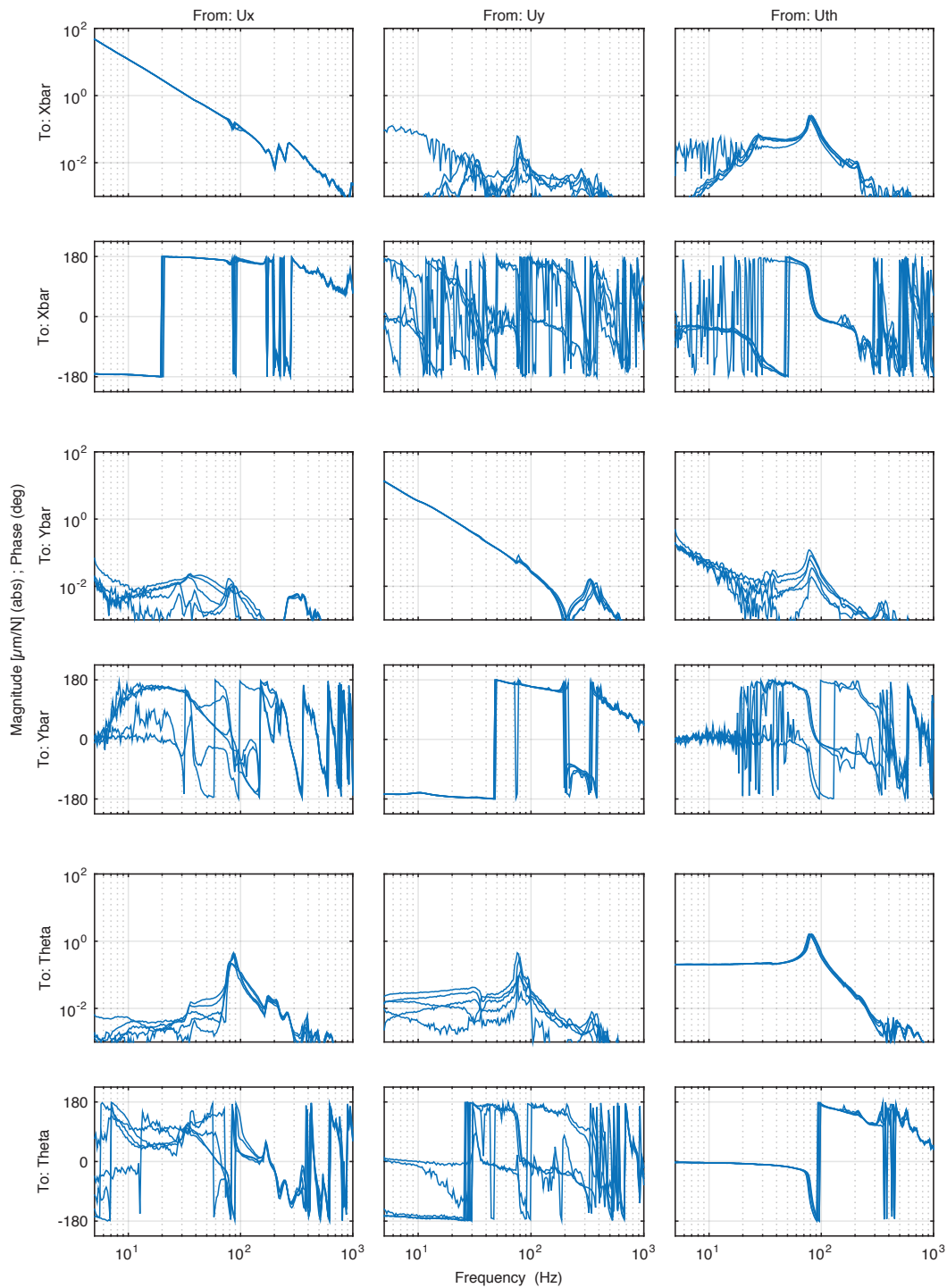


FIGURE 4. Transformed experimental frequency response measurements (displacement per force) from the AGS15000 now show much less coupling between the inputs and outputs. The responses from X and Y body forces are largely mass-dominated, and the response from differential motor forces to the bridge rotation shows a spring-like response at frequencies below the fundamental resonance.

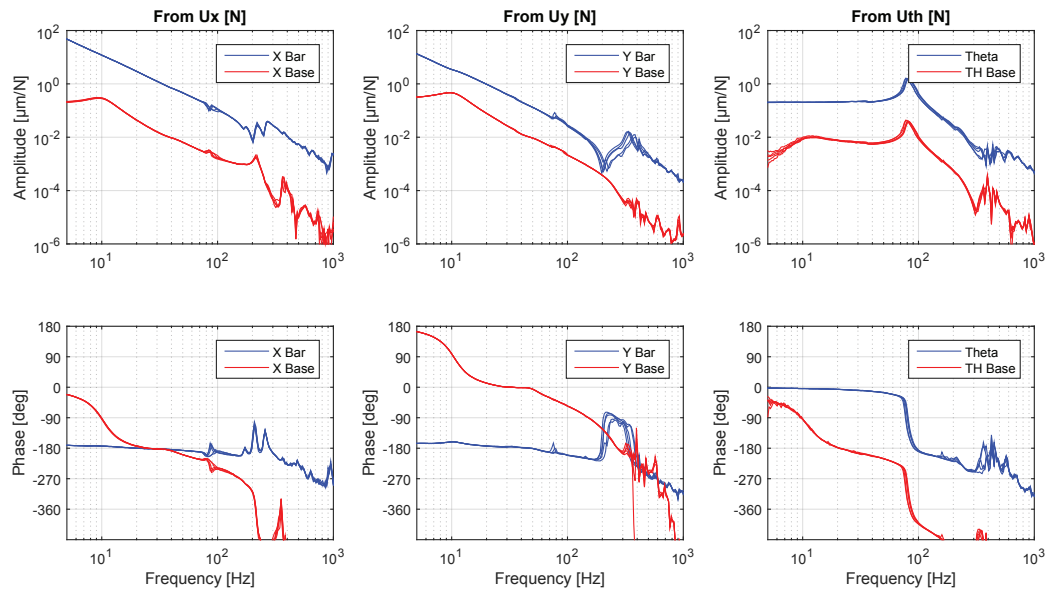


FIGURE 5. Additional sensors on the base allow measuring frequency responses of both the stage and base. The offset between the plots is due to the ratio of fixed to moving inertia.

REFERENCES

- [1] R. Hoogendijk, R. van de Molengraft, and M. Steinbuch, "Data-based control of lightweight high-precision motion systems," *Mikroniek*, vol. 52, no. 5, pp. 22–27, 2012.
- [2] D. J. Gordon and K. Erkorkmaz, "Precision control of a T-type gantry using sensor/actuator averaging and active vibration damping," *Precision Engineering*, vol. 36, no. 2, pp. 299–314, 2012.
- [3] I. García-Herreros, X. Kestelyn, J. Gomand, R. Coleman, and P.-J. Barre, "Model-based decoupling control method for dual-drive gantry stages: A case study with experimental validations," *Control Engineering Practice*, vol. 21, pp. 298–307, 2013.
- [4] C. Teo, K. Tan, S. Lim, S. Huang, and E. Tay, "Dynamic modeling and adaptive control of a H-type gantry stage," *Mechatronics*, vol. 17, no. 7, pp. 361–367, 2007.
- [5] J. Y. Yoon and D. L. Trumper, "Friction modeling, identification, and compensation based on friction hysteresis and Dahl resonance," *Mechatronics*, 2014.

INPUT SIGNAL FOR LEVITATION CAUSED BY VERTICAL VIBRATION OF A PIEZOELECTRIC ACTUATOR

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INTRODUCTION

Small factory products with high value are manufactured by highly performance production systems. An automated production system, an intelligent manufacturing system, and a flexible production system etc. are introduced. A small-scale production system using miniature robots, which was introduced in 1993, can save energy cost and production space [1].

Friction is the resistance that one surface of object encounters when moving over another surface. The friction force, which prevents a small part from moving, works on a surface. The smaller a product is, the larger the ratio of the surface area to volume is. Therefore, the friction force working on the small part is dominant. The friction should be controlled if the small-scale production system is concerned.

We have developed a lot of miniature robots used in the small-scale production system [2]. These robots use piezoelectric actuators (piezos) and electromagnets. The principle of the robots is based on an inchworm. The clamp of the electromagnet and the deformation of the piezo realize inchworm motion. Since the piezo deforms in several micron, the robots realize precise motion. The principle of an inchworm motion realizes wide working range.

These miniature robots move by the principle of an inchworm, and slide on a surface. The friction sometimes interferes with the motion of the inchworm-type miniature robots. Non-contact motion is desired.

Vertical vibration of a piezo causes vertical levitation, which was used in an inchworm type miniature robot [3]. The control signal which changed the levitation height was determined experimentally. Characteristics of input signals of the piezo used in the vertical levitation were not clear.

This paper describes levitation caused by the piezo which vibrates in the vertical direction. Input signals applied to the piezo and the vertical levitation of the piezoelectric levitation actuator is described.

LEVITATION MECHANISM

An actuator described in this paper is quite simple. The actuator which is a levitation mechanism is shown in Figure 1. The mechanism consists of a weight, a piezo, and a circular plate. They are attached by adhesive. The plate is 3 cm in diameter and 3 mm in thick. A stacked-type piezo (NEC-Tokin, AD050516, 20 mm long) is used. When the input voltage of 100 V_{DC} is applied to the piezo, it extends approximately 11 μ m. When the high-frequency voltage is applied to the piezo, it vibrates vertically. The magnitude of the deformation of the piezo varies according to the frequency of the input signal. The amplitude of the input voltage also changes the magnitude of the deformation of the piezo.

In our previous work, levitation mechanisms are connected, and they consist an inchworm which can move in the linear and rotational direction [3]. Horizontally-deformable piezos are inserted between the levitation mechanisms. The levitation mechanisms levitate alternately. While one levitation mechanism is levitating, the other levitation mechanisms are on a surface. The mechanisms which are not levitating does not move since the friction force keeps their positions. The levitating one is moved by the deformation of the horizontal piezo. The sequential motion realizes the inchworm motion. The rise time and fall time of the levitation mechanisms are not clear, since they show non-linear phenomena. Quick rise and fall are required to increase the velocity of the levitation type inchworm.

The squeeze film, which is generated underneath the circular plate by the vertical vibration of the piezo, lifts the levitation

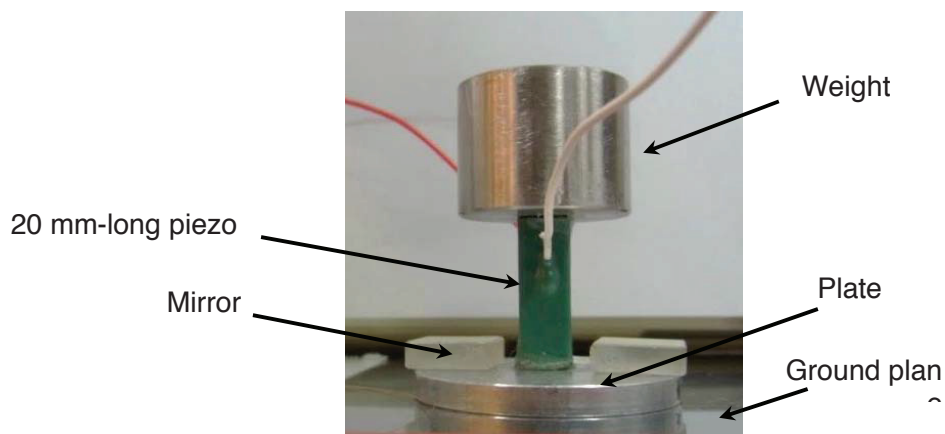


FIGURE 1. A mover (levitation mechanism) which levitates by the vertical vibration of a piezo.

mechanism. The mechanism is placed on a flat plane on a tilt stage. As shown in Figure 1, mirrors are fixed on the circular plate which are used for the levitation height measurement. The levitation height, which equals the thickness of the air film, is measured with an optical fiber displacement sensor.

The control signal is generated by an oscillator and is applied to the piezo through an amplifier. The levitation height, input voltage and current, phase difference between them, and power are measured, simultaneously. A power meter is used. The control frequency is 10.6 kHz, since the maximum levitation was obtained at that frequency when the 10 V peak-to-peak (Vpp) is applied. Since the levitation mechanism shows the nonlinear phenomenon, a mathematical model is not used in this paper. The characteristics are determined experimentally.

The vibration amplitude and frequency varies the thickness of the air film and the levitation height. When the DC voltage is applied to the piezo, friction works against the motion of the mover in the horizontal direction. When the AC voltage at the appropriate frequency, which is usually the resonant frequency, is applied to the piezo, the air film lifts the mover in the vertical direction and the friction does not work. The vertical levitation is larger than the static deformation of the piezo. In the experiment, the maximum static deformation of the piezo in the longitudinal direction is 6 μm . The maximum AC voltage applied to the piezo is 10 Vpp.

EXPERIMENTAL RESULTS AT CONSTANT FREQUENCY

When DC voltage is applied to the piezo, the piezo is charged statically. The piezo usually behaves like a capacitive load, when AC voltage is applied to the piezo. Voltage $v(t)$ and current $i(t)$ applied to the piezo is expressed by

$$v(t) = C \frac{di(t)}{dt}, \quad (1)$$

where C denotes the equivalent capacitance of the piezo. From equation (1), when the input voltage is sine wave, the input current must be cosine wave.

While levitating at 10.6 kHz, the levitation mechanism consumes energy. The measured voltage and current, which are applied to the levitating mechanism, are shown in Figure 2. As shown in Figure 2(a), the voltage and current oscillate in almost the same phase. Figure 2(b) shows the Lissajous figure of the input signals. Although Figure 2(b) does not show that the current and voltage are in the same phase, the phase difference is not $\pi/2$.

When the input voltage is 1 V, the mover does not float. The phase difference of the input voltage and current is about 70 degrees. However, the larger the input voltage is, the smaller the phase difference is. For example, when the input voltage is 2 V, the phase difference is about 30 degrees.

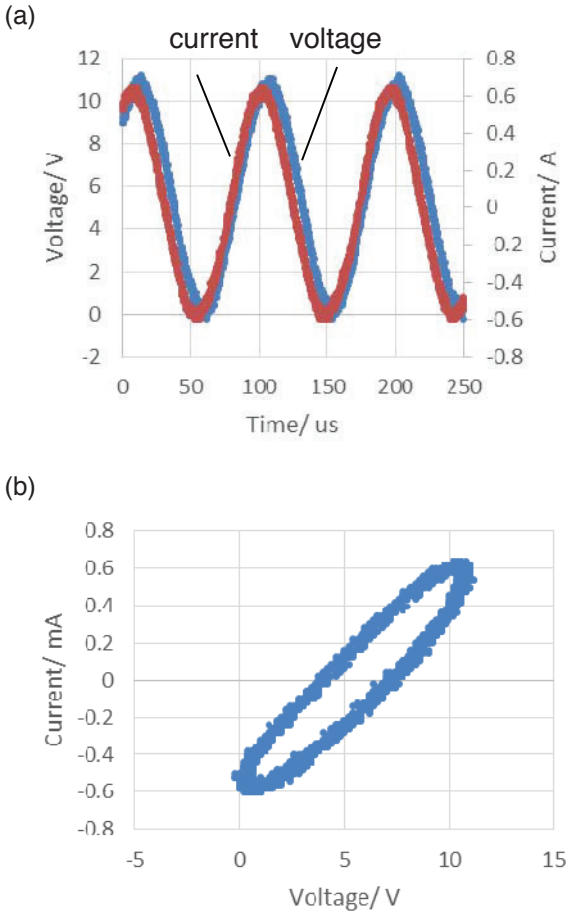


FIGURE 2. (a) Input voltage and current to the piezo of the levitation mechanism. (b) Lissajous figure of the input signal.

The levitation height (vertical displacement), input voltage, input current, power consumption, and phase difference between the voltage and current are measured simultaneously.

Figures 3-6 show the results. The frequency used in the experiments is 10.6 kHz at which the maximum levitation height is obtained. The input voltage and levitation height are plotted in Figure 3, and the input current and levitation height are plotted in Figure 4. They show the similar trend. For the voltage and current smaller than the threshold values, the levitation does not occur. In Figures 5 and 6, levitation height is expressed as a function of the power consumption and phase difference. These results are really important. By measuring the phase difference of the input signal and/or the power consumption, the levitation height can be estimated. These

results imply that the vertical displacement is estimated without using displacement sensors.

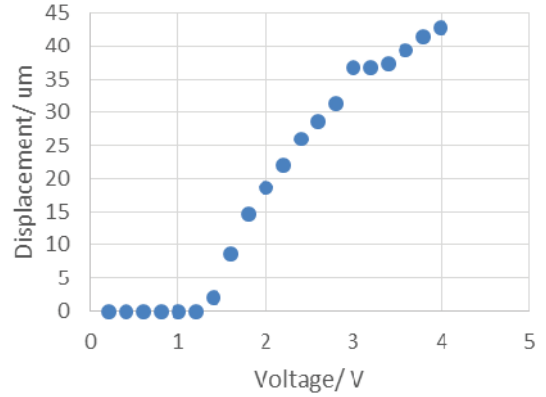


FIGURE 3. Levitation height vs. input voltage.

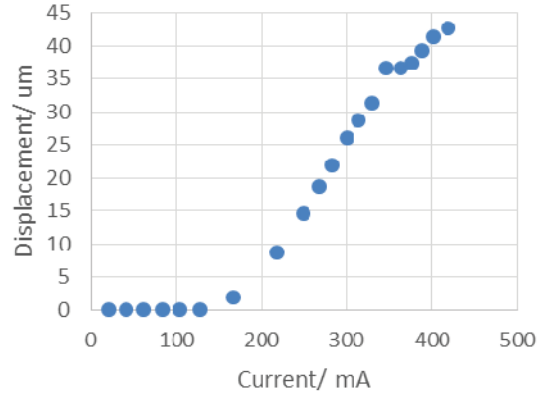


FIGURE 4. Levitation height vs. input current.

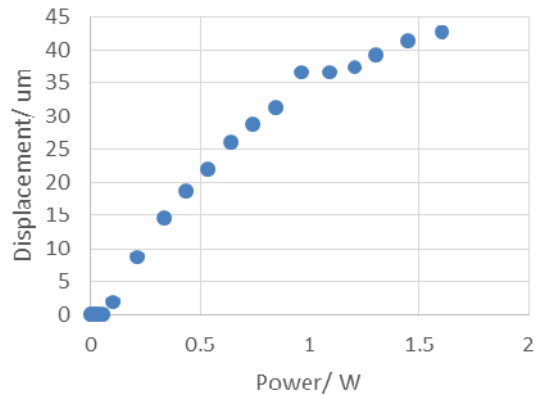


FIGURE 5. Levitation height vs. power consumption.

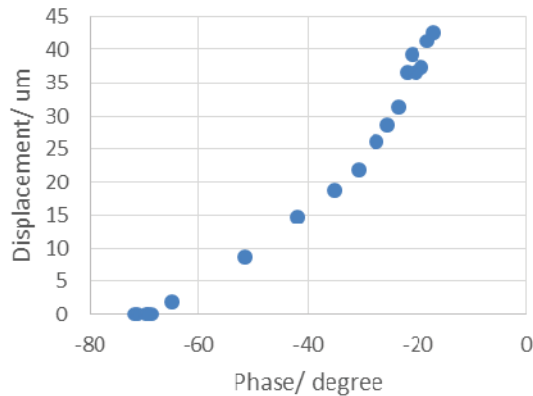


FIGURE 6. Levitation height vs. phase difference.

FREQUENCY CHARACTERISTICS

The levitation displacement, power consumption, and phase difference between the input voltage and current are measured as a function of frequency. The input voltage is 3.5 V.

Figures 7 and 8 show the results. The maximum levitation height, the maximum power consumption, and the highest peak of the phase difference are obtained at the equal frequency, which is 10.6 kHz. The levitation height which is controlled by changing the control frequency is estimated by measuring the power consumption and phase difference around the resonant frequency. The levitation height therefore can be estimated by the phase difference and the power consumption of the input signal applied to the piezo.

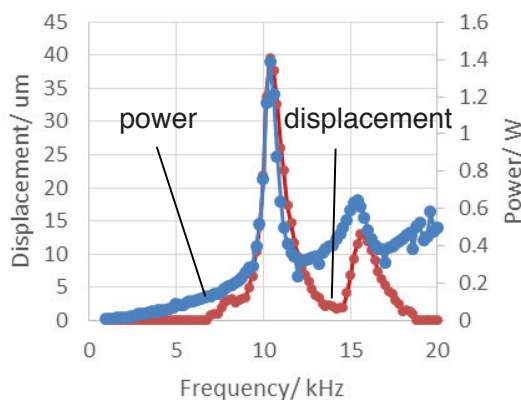


FIGURE 7. Levitation height and power consumption as a function of input frequency.

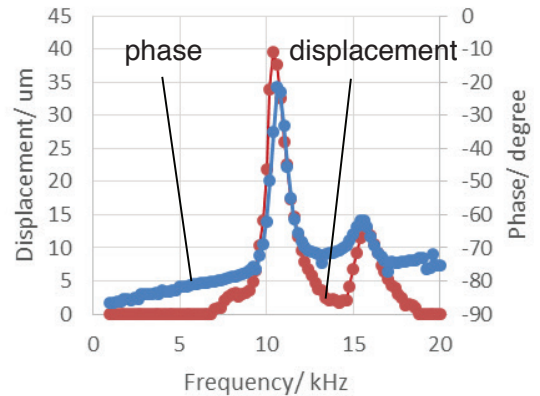


FIGURE 8. Levitation height and phase difference as a function of input frequency.

SUMMARY

This paper describes levitation caused by the piezo which vibrates in the vertical direction. Input signals applied to the piezo and the vertical levitation are measured. The power consumption and phase difference of the piezoelectric actuator have close relation to the levitation height. These results imply that the vertical levitation height can be estimated without using displacement sensors.

ACKNOWLEDGEMENT

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REFERENCES

- [1] Aoyama H, Nakano H, Hayakawa N, Sasaki A, Shimokohbe A. Miniature Robots for Ultra Precision measurement and Machining. International Progress in Precision Engineering. 1993; 275-289.
- [2] Torii A, Yamada T, Kamiya R, Ueda A, Doki K. An Inchworm-type Multi-DOF Stage. IEEJ Transactions on Electronics, Information and Systems. 2013; 133: 849-855. (in Japanese)
- [3] Torii A, Nishio M, Doki K, Ueda A. A 3-DOF Inchworm Using Levitation Caused by Vertical Vibration. IEEJ Transactions on Electronics, Information and Systems. 2014; 134: 1716-1723. (in Japanese)

INTERFEROMETRIC ACTIVE INERTIAL ISOLATION FOR EXTENDED STRUCTURES

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Abstract

This article presents an active vibration isolation strategy combining centralized interferometric inertial control and decentralized force control loops. This innovative approach is designed to comply with the stringent requirements of particle detectors where coils and magnets cannot be used. The technology and control approach developed is also of interest for potentially improving the performance of isolators used in gravitational wave detectors such as LIGO, whose active platforms performance is limited by inertial sensor noise at low frequencies, and control bandwidth at high frequencies.

1 INTRODUCTION

Effective isolation from ground vibration is required for many applications including lithography machines, atomic force microscopy, medical imaging, and large instruments dedicated to experimental physics [1]. Among them, the stabilization of the electromagnets of future compact linear particle collider (CLIC) is particularly challenging [2-4]. It is estimated that the RMS value above 4 Hz of the last electromagnet vertical displacement should not exceed 0.15 nm. Beyond these stringent requirements, several additional constraints must be adequately addressed. They are related to the extended shape of the structure, the limited stability of the floor on which they are mounted, and limitations on the technology of the sensors and actuators which could couple with the electromagnets.

A dedicated control approach is proposed in this paper to meet these requirements. It includes key features to comply with the aforementioned constraints. Firstly, the instrumentation does not use any coil, magnet and elastomer, in order to be compatible with both magnetic field and radiation. Secondly, the mechatronic architecture is dedicated to support long and extended objects. Thirdly, the controller relies on a fusion of interferometric inertial sensor and force sensor, which offers unique stability properties to the closed loop system, and a high robustness against plant spurious resonances (this is especially suited as the electromagnets will be mounted on an unstable support).

While this control strategy is specifically designed for the CLIC, several of the features and components of the instrumentation are also of potential interest for the LIGO in-vacuum isolators, whose performance is currently partly limited by the sensor noise of its vertical inertial sensors at low frequency, and by the control bandwidth at high frequency [5, 6].

Section two presents the interferometric sensor (NOSE) used for the inertial control; section three presents the active isolation system, including the mechatronic architecture, the control strategy and preliminary experimental results. Section four gives the conclusion.

2 INTERFEROMETRIC SENSOR: NOSE

Fig. 1. Shows a picture of the NOSE prototype.

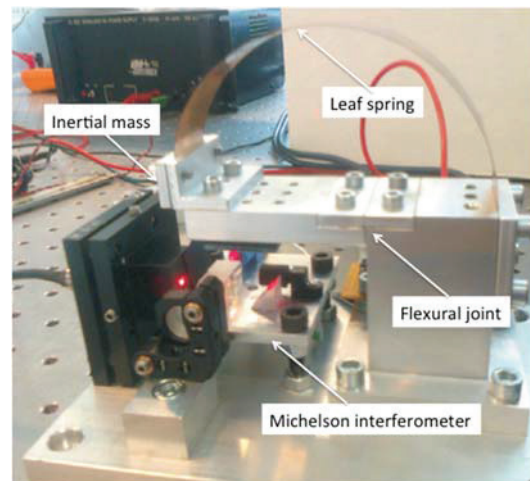


FIGURE 1. Picture of the interferometric inertial sensor NOSE.

The mechanical part consists of a horizontal pendulum, connected to a rigid frame through a flexural joint, made of CuBe alloy. A leaf spring, made of the same alloy, is used to adjust the equilibrium position of the inertial mass and balance the gravity. The oscillator is characterized by an inertial mass $m=0.055\text{kg}$, a main resonance frequency $f_0=6\text{Hz}$ (tunable) and spurious resonances above 100 Hz. NOSE does not contain any loaded coil, which was found to offer several advantages, including compatibility with magnetic environments and a low thermal noise in the suspension (Brownian motion) [7].

In order to measure the relative displacement between the inertial mass and the support, a sensor based on a Michelson interferometer has been developed and adapted to enable the measurement of both quadratures of the signals as described in [8].

Figure 2 shows the experimental amplitude spectral density of the interferometer noise (blue dashed-dotted line), expressed in physical units of $[\text{m}/\sqrt{\text{Hz}}]$. It has been obtained in two steps, while the motion of the inertial mass is restrained.

In the first step, we manually slightly moved one mirror in order to obtain the parameters of the Lissajous figure for the calibration.

In a second step, we have recorded the photodiode signal without disturbing the interferometer, combined with the parameters obtained from the first step in order to express the photodiode signals in displacement units, which is considered to be the sensor noise. It results in a resolution of $3 \text{ pm}/\sqrt{\text{Hz}}$ above 1 Hz, and $20 \text{ pm}/\sqrt{\text{Hz}}$ above 0.3 Hz [6]. Including the inertial mass dynamics, the resolution remains $3 \text{ pm}/\sqrt{\text{Hz}}$ above 4 Hz (red line in Fig.2).

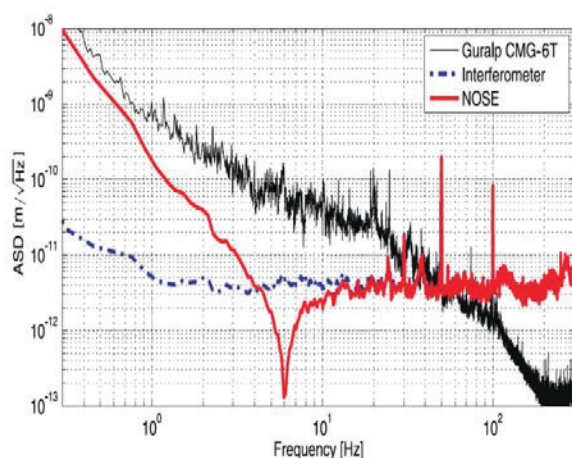


FIGURE 2. Comparison of sensor experimental resolution: Guralp CMG-6T (black line, obtained

from two sensors side by side placed in a quiet environment); resolution of our NOSE interferometer, measured by blocking the inertial mass; estimated resolution of NOSE (red line) [9].

3 ACTIVE ISOLATION SYSTEM

3.1 Mechatronic architecture

In order to control the six degrees-of-freedom of a sensitive equipment, at least six actuators are required. Depending on the feedback control objective, the supports are either oriented parallel/perpendicular to the gravity [10, 11], or inclined in the manner of a Stewart platform [12]. For the final focus CLIC quadrupoles, a high authority is required in the vertical direction and a weaker authority in the lateral, yaw and pitch directions.

No authority is required in roll and longitudinal directions. Due to the extended shape of the electromagnet, it has been decided to use eight supports, as shown in green in Fig. 3: Four in the vertical direction (two at each end), and four in the lateral direction, with a small angle to restrict the longitudinal motion.

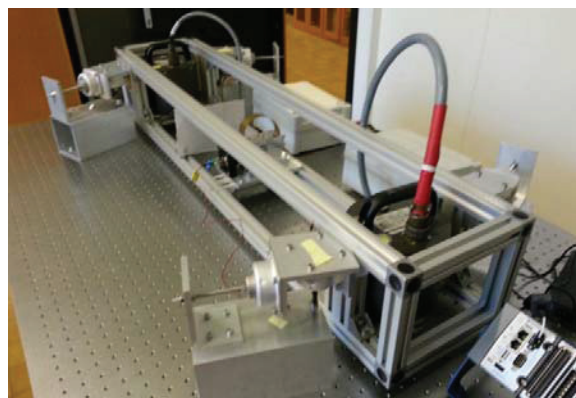


FIGURE 3. Picture of the active isolation system

3.2 Control strategy

A first concept was studied, where each leg was made of an actuator mounted in series with an inertial sensor as described in [13]. The idea was to collocate actuator/sensor pairs and use SISO decentralized controller in each mount. In this configuration, each actuator compensated the absolute motion of the quadrupole, measured in the direction of the actuator. However, this idea was rapidly withdrawn due to practical difficulties: it is not straightforward to mount an inertial sensor in series with an actuator, the configuration does not guarantee the stability of the feedback loops, and it is difficult to develop both small and sensitive inertial sensors.

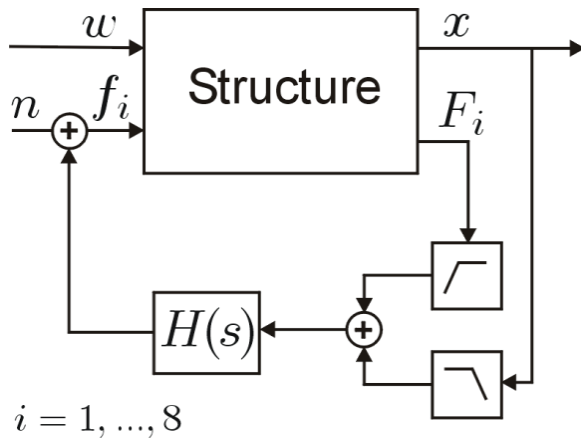


FIGURE 4. Closed loop bloc diagram. The strategy combines a centralized inertial control at low frequency, and decentralized force control loops in each active leg.

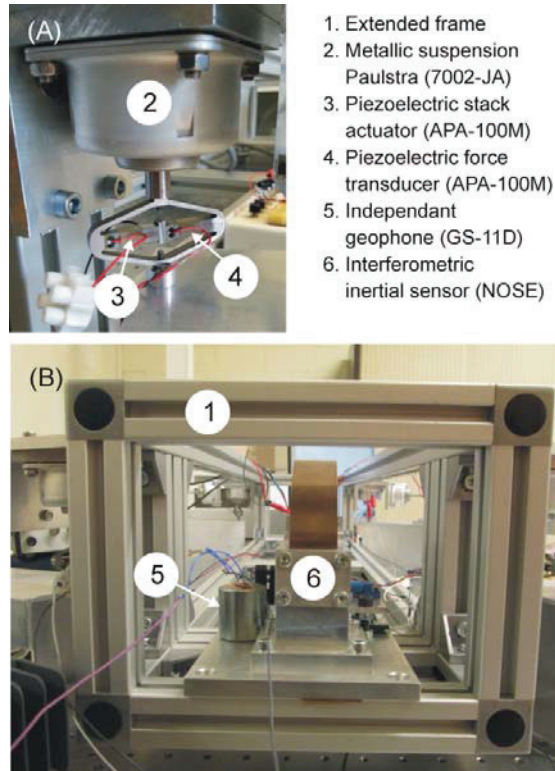


FIGURE 5. Picture of the experimental set-up. (A) One active leg, constituted of a dual pair of piezoelectric actuator/sensor, in series with a metallic suspension. (B) Picture of the interferometric inertial sensor (NOSE) used in the feedback loop, and of the witness geophone used to assess the closed loop performance.

For these reasons, we have decided to use high resolution inertial sensor [8] (NOSE presented in section 2) at low frequency for centralized inertial

control and fusion the inertial sensor at high frequency with decentralized force feedback loops in each mount, as shown in Fig. 4, where $i=1,\dots,8$. The method has been published in [14].

In the experimental set-up, each active mount is made of a piezoelectric stack actuator APA100M from Cedrat Technologies, mounted in series with a metallic suspension Paulstra 7002-JA to offer passive isolation at high frequency (Fig.5A). This configuration offers the interesting property to be completely compatible with particle accelerator environments. Each piezoelectric stack actuator is further divided in two parts: one is used as actuator, whereas the other one is used as force sensor, the assembly making a dual truly collocated actuator/sensor pair.

3.3 Experimental results

The control strategy has been implemented in a Dspace DS1103 at 5kHz. For the time being, only the vertical direction has been tested. In order to remain above the noise floor of the witness geophone, random noise (n) is injected in the actuators, and the equipment motion is measured by a geophone (x).

Figure 6 shows the transmissibility between n and x when the controller is turned off (T_{off}) and turned on (T_{on}). The feedback operation reduces the transmitted motion by three orders of magnitudes. The performance is limited by the geophone noise, but not by feedback stability issues.

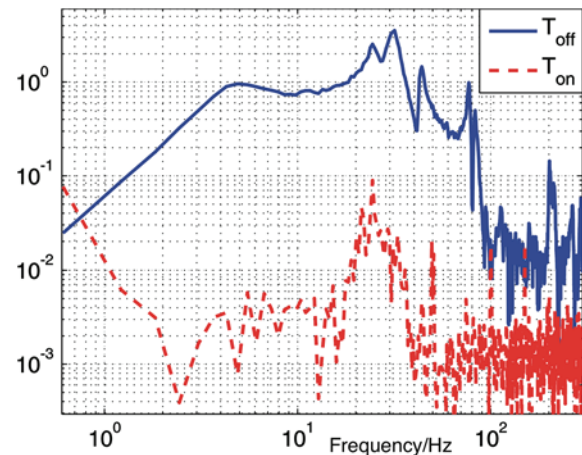


FIGURE 6. Vertical transmissibility between the noise (n) injected in the actuator and the structural vibrations measured by the witness geophone (x), integrated, and normalized, when the controller is turned off (T_{off}) and turned on (T_{on}).

4 CONCLUSION

The paper has presented preliminary experimental results of an active vibration isolation stage, dedicated to support extended payloads. To the best of the author's knowledge, this is the first active isolation system which does not contain any coil, magnet or elastomer, which makes it fully compatible with accelerator environments. Using the proposed strategy, preliminary results have shown that transmitted motion can be reduced by three orders of magnitude (60 dB) when the controller is turned on. In a future work, the closed loop performance in the lateral direction will be tested as well.

Though the inertial sensor do not meet sensor noise requirements to potentially improve the LIGO active platforms, the prototype lays out technology of interest for a vacuum compatible and magnetic insensitive vertical interferometric sensor. Additionally, the active control results demonstrate that the blend of inertial sensors and force sensors permit to achieve very large bandwidth, which is also of potential interest for the enhancement of active isolators used in gravitational wave detectors.

5. ACKNOWLEDGMENTS

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REFERENCES

- [1] Matichard, F., et al., *Seismic isolation of Advanced LIGO: Review of strategy, instrumentation and performance*. Classical and Quantum Gravity, 2015. **32**(18): p. 185003.
- [2] Collette, C., S. Janssens, and D. Tshilumba, *Control strategies for the final focus of future linear particle collider*. Nuclear Instruments and Methods in Physics Research Section A: Accelerators, Spectrometers, Detectors and Associated Equipment, 2012. **684**: p. 7-17.
- [3] Collette, C., et al., *Nano-motion control of heavy quadrupoles for future particle colliders: An experimental validation*. Nuclear Instruments and Methods in Physics Research Section A: Accelerators, Spectrometers, Detectors and Associated Equipment, 2011. **643**(1): p. 95-101.
- [4] Le Breton, R., et al., *Nanometer scale active ground motion isolator*. Sensors and Actuators A: Physical, 2013. **204**: p. 97-106.
- [5] Matichard, F., et al., *Advanced LIGO two-stage twelve-axis vibration isolation and positioning platform. Part 1: Design and production overview*. Precision Engineering, 2015. **40**: p. 273-286.
- [6] Matichard, F., et al., *Advanced LIGO two-stage twelve-axis vibration isolation and positioning platform. Part 2: Experimental investigation and tests results*. Precision engineering, 2015. **40**: p. 287-297.
- [7] Saulson, P.R., *Thermal noise in mechanical experiments*. Physical Review D, 1990. **42**(8): p. 2437.
- [8] Zumberge, M., et al., *An optical seismometer without force feedback*. Bulletin of the seismological society of America, 2010. **100**(2): p. 598-605.
- [9] Collette, C., et al., *Prototype of interferometric absolute motion sensor*. Sensors and Actuators A: Physical, 2015. **224**: p. 72-77.
- [10] Vervoordeldonk, M. and H. Stoutjesdijk. *Recent developments, a novel active isolation concept*. in *6th euspen International Conference, Baden bei Wien*. 2006.
- [11] Beard, A.M., D.W. Schubert, and A.H. von Flotow. *Practical product implementation of an active/passive vibration isolation system*. in *SPIE's 1994 International Symposium on Optics, Imaging, and Instrumentation*. 1994. International Society for Optics and Photonics.
- [12] Stewart, D., *A platform with six degrees of freedom*. Proceedings of the institution of mechanical engineers, 1965. **180**(1): p. 371-386.
- [13] Anderson, E.H., J.P. Fumo, and R.S. Erwin. *Satellite ultraquiet isolation technology experiment (SUITE)*. in *Aerospace Conference Proceedings, 2000 IEEE*. 2000. IEEE.
- [14] Collette, C. and F. Matichard, *Sensor fusion methods for high performance active vibration isolation systems*. Journal of sound and vibration, 2015. **342**: p. 1-21.

A MULTI-AXIS MOTION CONTROL SOLUTION OF LINEAR ULTRASONIC MOTORS FOR MICRO CNC MACHINE TOOLS - POWERED BY INDUSTRIAL ETHERNET TECHNOLOGY (ETHERCAT)

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INSTRUCTIONS

Linear ultrasonic motors have great advantages in high accuracy, self-locking, anti-EML, etc., thus are suitable to be used as a motion stage in a micro machine tool[1]. However, there are currently very seldom reports on using linear ultrasonic motors for a machine tool application. One of the most important reasons why the multi-axis ultrasonic motor stage is hard to be used in a high-precision machine tool is that the control technology for ultrasonic motors is not mature enough for NC tasks. The high precision feed motion implies high contour accuracy (rather than point-to-point accuracy), superfine interpolation, large look-ahead buffer, hard real-time ability, axes group control, high synchronization, etc. Unfortunately, the recent motion control vendors for ultrasonic motors have few considerations on CNC applications.

That are several approaches to solve the above mentioned control problem. Obviously, it is not reasonable to develop another CNC system from scratch. Another brilliant choice is to introduce the Open Architecture CNC (Open-CNC) System from the conventional CNC machine field [2]. A open-architecture CNC system can be used for ultrasonic motors, if only the ultrasonic motor servo has a compatible control interface. Therefore, the critical task is to develop a ultrasonic motor servo which can be controlled by a open-architecture CNC system.

This article introduces the servo controller technology for multi-axis motion stages driven by linear ultrasonic motors. The servo controller presented in this article is developed as a standalone module, and is the first reported servo controller of ultrasonic motors powered by an industrial Ethernet technology (EtherCAT) [3].

Firstly, The designs of amplifier and control circuits are proposed. Secondly, the EtherCAT communication interface is developed on the controller. In together, those three modules compos-

ite the main body of the servo controller. Thirdly, a five-axis TRIPOD motion stage is developed for micro machine tool usage, which verifies the feasibility of the developed servo controller.

INTRODUCTION OF L1-B2 ULTRASONIC MOTOR

The L1-B2 (first-order elongating and second-order bending modes) ultrasonic motor is one of the most frequently used linear ultrasonic motors. It is easy to make and convenient to drive[4]. The working principle is illustrated in Fig. 1. The resonant frequencies of the first-order elongating and the second-order bending modes are designed close to each other, thus the excitation voltage can generate vibrations of both modes. The combination of vibrations will at last generate an elliptic motion on the tip, which subsequently drive the mover forwards or backwards via friction effects.

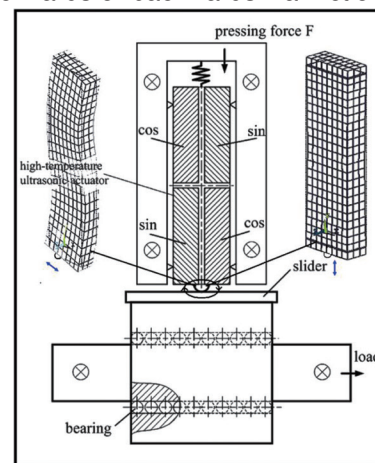
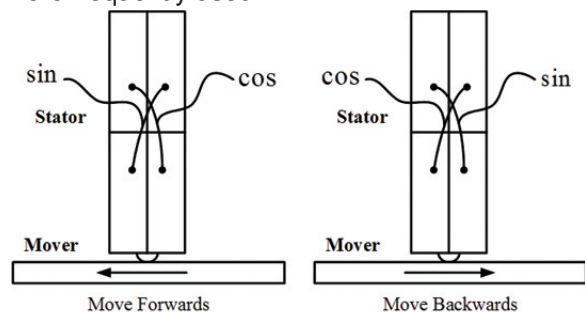


FIGURE 1. The working principle of a L1-B2 linear ultrasonic motor.

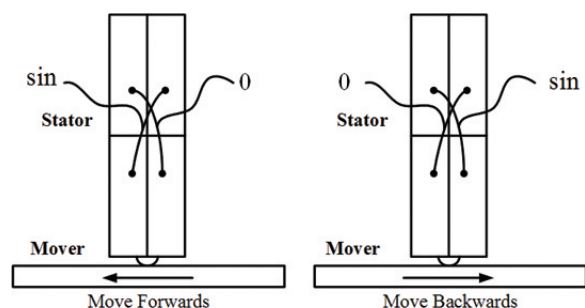
In order to drive the L1-B2 motor, a high AC voltage with its amplitude up to 800V~1000V needs to be provided [5]. Therefore, a special driving circuit has to be designed, and a corresponding controller which generates the necessary PWM waves, receives the encoder feedback and performs the PID algorithm has to be developed.

DRIVE CIRCUIT OF ULTRASONIC MOTOR

There are generally two driving modes for a L1-B2 ultrasonic motor, single-phase and double-phase modes, as shown in Fig. 1. Comparing to the double-phase mode, the single-phase driving mode needs only one resonant high voltage rather than two, hence is relatively simpler. Therefore, practically the signal-phase mode is more frequently used.



(a) Drive a L1-B2 ultrasonic motor using a single-phase mode



(b) Drive a L1-B2 ultrasonic motor using a double-phase mode

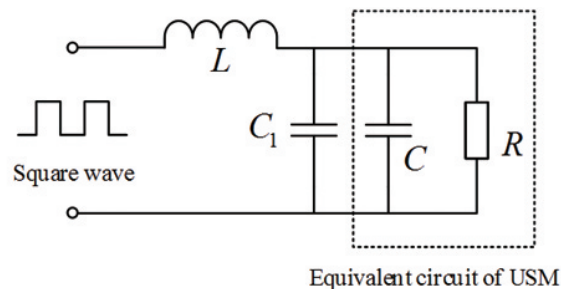
FIGURE 2. Driving modes of a L1-B2 ultrasonic motor.

In order to generate the high AC voltage, the LC resonant circuit is used. With carefully calibrated parameters of the inductor and capacity, the high resonant voltage can be converted from a 24V DC voltage source. The resulted AC voltage has a frequency of circa 50 KHz, and an amplitude between 400V ~ 800V[6]. Fig. 3(a) presents the resonant driving circuit together with the ultrasonic motor's equivalent circuit, and Fig. 3(b) presents the H-bridge circuit used for generating the square wave power voltage. Fig. 4 is the driver circuit prototype of L1-B2 linear ultrasonic motor designed in this paper.

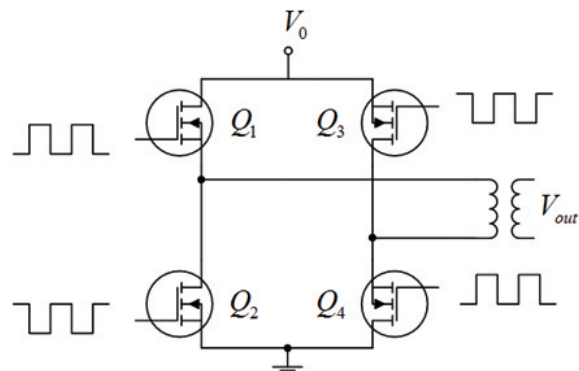
CONTROL CIRCUIT OF ULTRASONIC MOTOR

Generally, the control of a linear ultrasonic motor mainly has the following characteristics [6]:

(1)The controlling feature strongly depends on



(a) LC resonant driving circuit and ultrasonic motor's equivalent circuit



(b) H-bridge circuit for generating the square wave from PWM waves

FIGURE 3. LC resonant and H-bridge circuit for driving a ultrasonic motor.

the position of the linear ultrasonic motor.

(2)The curve between the velocity and the duty ratio has a significant dead zone.

(3)Positive and negative motion characteristics are not consistent.

In order to compensate the nonlinearity of the control system, so that the control algorithm can be simplified, the piecewise compensation should be conducted into the controller. There are two kinds of nonlinear compensations need to be considered.

(1)Compensation of the position varying dead zones

The compensation values, which can be looked up from a table of positive and negative voltages, could be directly added into the control variable for dead-zone compensation.

(2)Compensation of positive and negative inconsistency

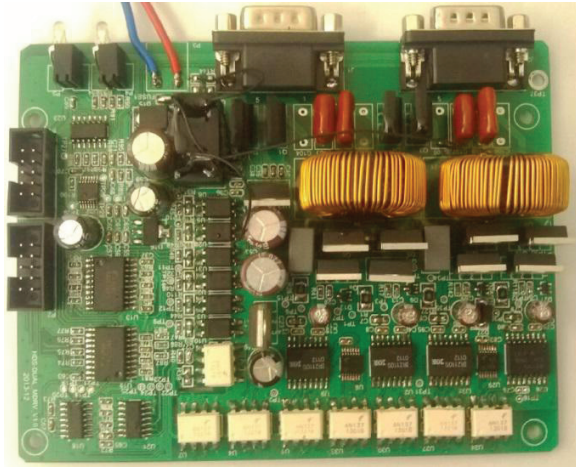


FIGURE 4. Linear ultrasonic motor driver circuit.

The inconsistency of forward and backward motions in a linear ultrasonic motor can cause difficulties in tuning the PID parameters for speed and position control tasks. Therefore, two slope adjusting functions are employed to correct the positive and negative motion characteristics.

Both of the compensations together with a PID controller are implemented in a Cortex-M3 ARM chip, as shown in Fig. 5.

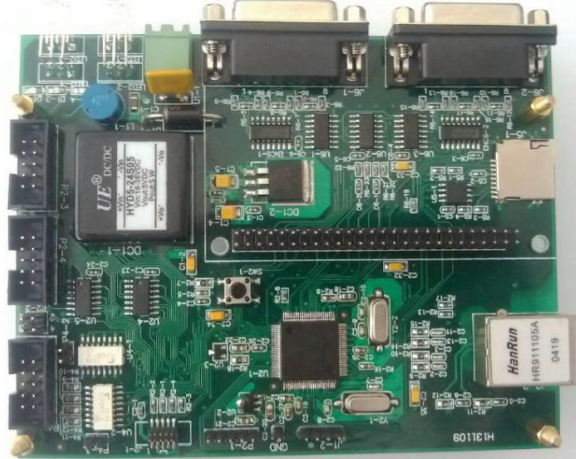


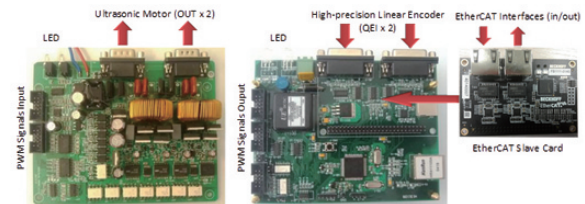
FIGURE 5. Linear ultrasonic motor control circuit.

MULTI-AXIS SOLUTION

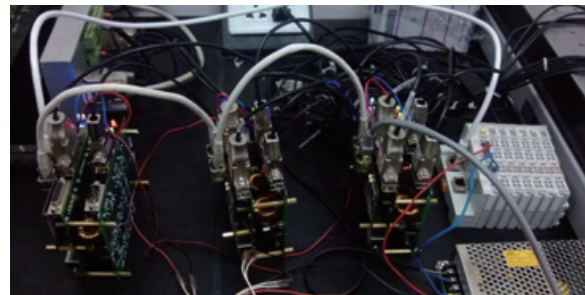
In the conventional industrial control field, the industrial Ethernet technology has greatly improved the development of open-architecture control systems. In addition, the industrial Ethernet solution is also suitable for multi-axis control. In order to provide the multi-axis control functionality

for piezoelectric motors, it is feasible and efficient to extend the industrial Ethernet technology to the piezoelectric field. In this paper, a ultrasonic motor servo controller with EtherCAT interface was developed, which also makes it possible for integrating the ultrasonic motor into the control network of conventional servo motors.

The correspondingly developed control circuits for linear ultrasonic motors are presented in Fig. 6.



(a) An integrated servo controller with the EtherCAT interface



(b) The multi-axis control solution with industrial Ethernet

FIGURE 6. The integrated servo controller and the multi-axis solution of ultrasonic motors powered by EtherCAT

TRIPOD STAGE CONTROL

Using the developed servo controller, a Tripod stage (Fig. 7) driven by five linear ultrasonic motors can be controlled in for micro machine tool applications. The Tripod stage consist of one X-Y serial stage and a 3-RPS parallel stage [7]. Accordingly, the forward and inverse kinematic algorithms are implemented in the master controller. Experiments shows that the servo controller with EtherCAT interface works successfully for linear ultrasonic motors. By introducing the powerful industrial Ethernet technology into the ultrasonic motor controller, can not only save a lot of developing work, but also it will bring many advanced control performance, such as shorter control period, higher synchronization, and hard real-time.

ACKNOWLEDGMENTS

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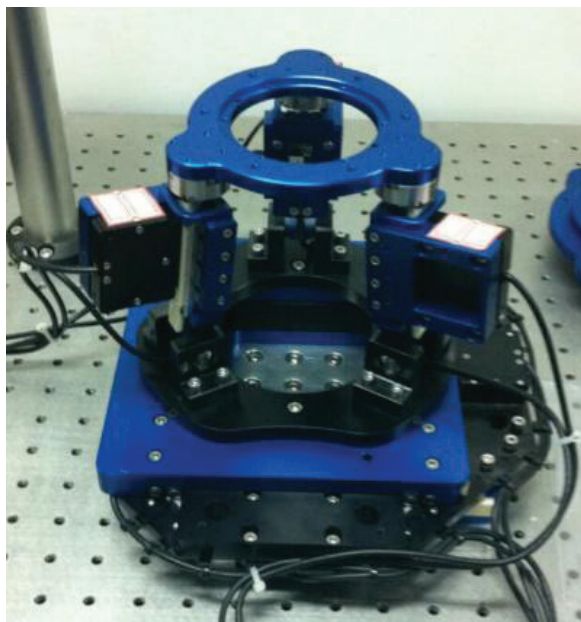


FIGURE 7. Tripod motion stage driven by linear ultrasonic motors.

China. (Project NO.: 51505020)

REFERENCES

- [1] Zhao CS. Ultrasonic motors technologies and application. Beijing, China: Science Press; 2007.
- [2] Suh SH, Kang SK, Chung DH, Stroud I. Theory and Design of CNC Systems. London, England: Springer-Verlag London Limited; 2008.
- [3] EtherCAT Technology Group, www.ethercat.org; 2015.
- [4] Li XT, Chen JG, Chen ZJ, X DS. A high-temperature double-mode piezoelectric ultrasonic linear motor. *Applied Physics Letters*. 2012;101(072902):1–5.
- [5] Bekiroglu E. Ultrasonic motors: Their models, drives, controls and applications. *Journal of Electroceramics*. 2007;20(3-4):277–286.
- [6] Shi J. Motion control theory and technology of ultrasonic motor. Beijing, China: Science Press; 2011.
- [7] Gallardo J, Orozco H, Rico J. Kinematics of 3-RPS parallel manipulators by means of screw theory. *The International Journal of Advanced Manufacturing Technology*. 2008;36(5-6):598–605.

Research on improving the thermal stability of hydrostatic guideways structure by realizing the homogenization of the temperature distribution

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INTRODUCTION

This paper proposed a new structure of closed oil hydrostatic guideways to realize the equalization of the distribution of temperature. Oil holes are set on the foundation of this new structure to reduce the thermal deformation and form error caused by uneven distribution of temperature, improve the thermal stability of hydrostatic guideways. According to this principle, the temperature mode of closed oil hydrostatic guideways is set up. By the calculation of finite element method (FEM), the oil holes on the guideways foundation are verified to have a significant effect on the homogenization of temperature distribution. The simulation results show that along with the increase of the oil hole diameter, the foundation had twice higher homogenizing temperature than that is without oil holes. The thermal deformation and form error follow the same trend with the temperature distribution, The thermal deformation and form error are reduced nearly 50%. This research for thermal stability of hydrostatic guideways provides a theoretical and technical support for hydrostatic guideways design.

CLOSED OIL HYDROSTATIC GUIDEWAYS WITH

OIL HOLES

Thermal errors of machine tools caused by temperature are related to many factors in the machining process. As the largest error source in the precision machine, thermal errors account for about 70% of the total errors [1-2]. Therefore, reducing the heat sensitivity of machine tools is the important requirement for machine design to improve the heat stability of machine tools. Generalized approaches to reduce the thermal deformations of machine critical components are summed up to the following categories, heat reduction, heat source isolation, cooling system and so on [3-10]. The power loss about hydraulic oil between parallel planes is the largest heat source on the closed oil hydrostatic guideways. Due to the structure of the closed oil hydrostatic guideways, thermal transmission from heat sources to critical components is inevitable. This paper proposed a structure of closed oil hydrostatic guideways with oil holes to realize the equalization of the distribution of temperature, reduce thermal deformation and form error caused by uneven distribution of temperature, and improve the thermal stability of hydrostatic guideways.

FIGURE 1. shows a typical closed oil hydrostatic guideways, slider covering rail, bearings placed at lateral region and top region of guideways. Then, in the machining process, temperature of outer region is higher than inner region, temperature of top region is higher than bottom. Guideways foundation is the critical component which has a decisive effect on precision of guideways. Deformation caused by uneven distribution of temperature directly impact on precision of guideways. To realize the homogenization of temperature distribution, oil holes interconnecting with each other are placed in the guideways foundation, to collect hydraulic oil back to oil tank through the same oil return pipe. FIGURE 2. shows a closed oil hydrostatic guideways with oil holes.

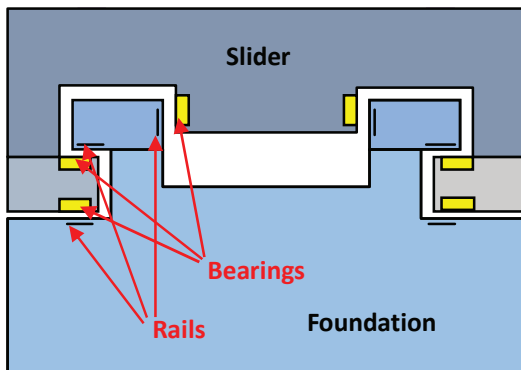


FIGURE1. A TYPICAL CLOSED OIL HYDROSTATIC GUIDEWAYS

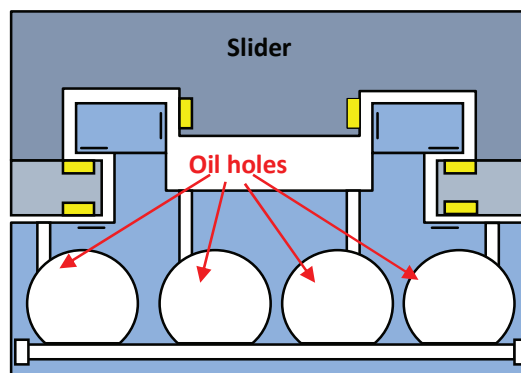


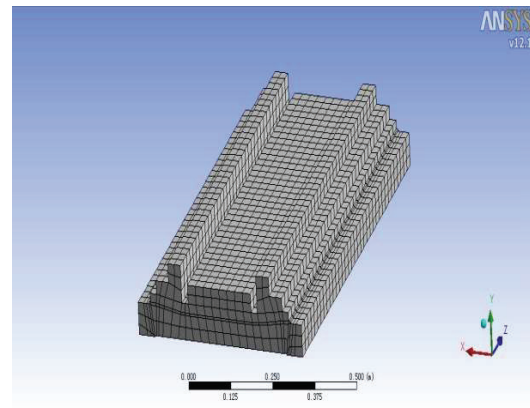
FIGURE2. A CLOSED OIL HYDROSTATIC GUIDEWAYSWITH OIL HOLES

FINITE ELEMENT MODE OF GUIDEWAYS FOUNDATION

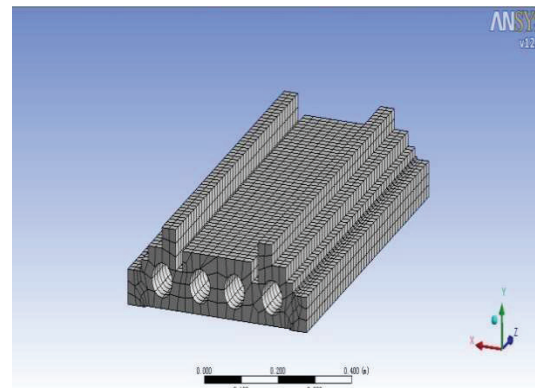
Temperature distribution and thermal deformation of guideways foundation are calculated by FEM. FIGURE.3 shows the finite element mode of guideways foundation.

The convective heat transfer coefficient h of heat transfer surfaces can be acquired through following formula (1):

$$h = \frac{N_{\mu} \cdot k}{l} \quad (1)$$



(1) GUIDEWAYS FOUNDATION WITHOUT OIL HOLES



(2) GUIDEWAYS FOUNDATION WITH OIL HOLES
FIGURE.3 FINITE ELEMENT MODE OF GUIDEWAYS FOUNDATION

RESULTS AND DISCUSSION

Influence Of Oil Holes On The Homogenization Of

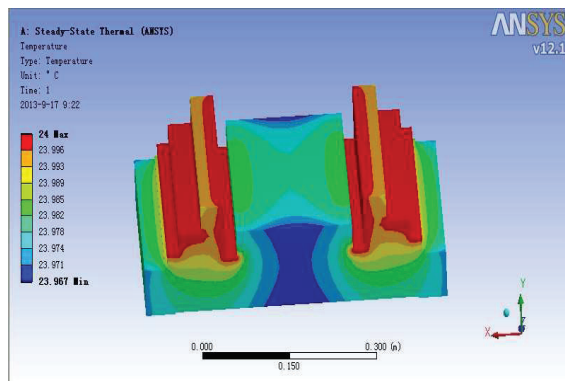
Temperature Distribution

The initial temperature(T_i) of guideways is 22°C. Due to continuous feeding of guideways slider, hydraulic oil spreads over guideways rails, temperature rise(ΔT) of all surfaces that hydraulic oil flows through is considered identical. Temperature distribution of guideways foundation is calculated with finite element method analysis in condition1-5. Condition1 is set as $T_i=22^\circ\text{C}$, $\Delta T=2^\circ\text{C}$, Condition2 is set as $T_i=22^\circ\text{C}$, $\Delta T=4^\circ\text{C}$, Condition3 is set as $T_i=22^\circ\text{C}$, $\Delta T=6^\circ\text{C}$, Condition4 is set as $T_i=22^\circ\text{C}$, $\Delta T=8^\circ\text{C}$, Condition5 is set as $T_i=22^\circ\text{C}$, $\Delta T=10^\circ\text{C}$.

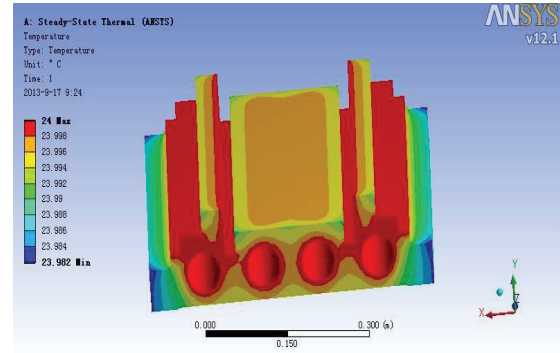
Maximum temperature difference (f) of guideways foundation is used to estimate the homogenization of temperature distribution, formula(2).

$$f = T_{\max} - T_{\min} \quad (2)$$

FIGURE.4 shows the influence of oil holes on the temperature distribution of guideways foundation. Temperature distribution of guideways foundation without oil holes appears that outer region is higher than inner region, top region is higher than bottom, which is discussed above. Temperature distribution of guideways foundation with oil holes is improved significantly.



(1) TEMPERATURE FIELD OF GUIDEWAYS FOUNDATION WITHOUT OIL HOLES



(2) TEMPERATURE FIELD OF GUIDEWAYS FOUNDATION WITH OIL HOLES
FIGURE4. INFLUENCE OF OIL HOLES ON THE HOMOGENIZATION OF TEMPERATURE DISTRIBUTION

FIGURE 5. shows the influence of oil hole diameter on maximum temperature difference (f) of guideways foundation. Under the same temperature rise, along with the increase of oil hole diameters, maximum temperature difference of guideways foundation with oil holes is decreased, it can be reduced more than half, especially compared to guideways foundation without oil holes. Under the different temperature rise, the effect trend of oil hole diameter on maximum temperature difference shows consistency.

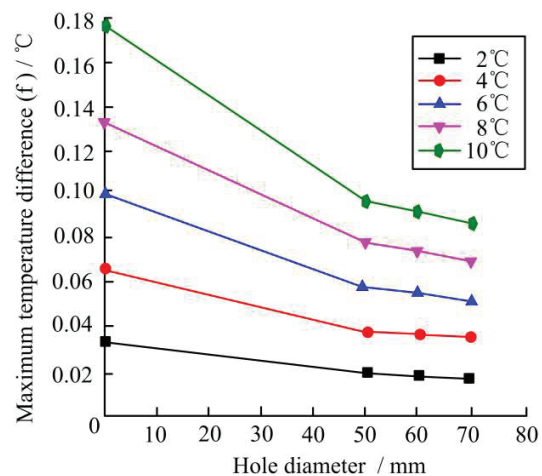


FIGURE5. INFLUENCE OF OIL HOLE DIAMETER ON MAXIMUM TEMPERATURE DIFFERENCE OF GUIDEWAYS FOUNDATION

Influence Of Oil Holes On The Thermal Deformation And Form Error

FIGURE 6. shows the influence of oil holes on thermal deformation of guideways rail in vertical direction. Thermal deformation of guideways rail in vertical direction displays large in the middle and little in the end to have the whole rail uplifted. Under the same temperature rise, and along with the increase of oil hole diameter, thermal deformation of guideways rail is decreased, it can be reduced by one-third, especially compared with guideways foundation without oil holes. Under the different temperature rise, the effect trend of oil hole diameter on thermal deformation of guideways rail in the vertical direction shows consistency.

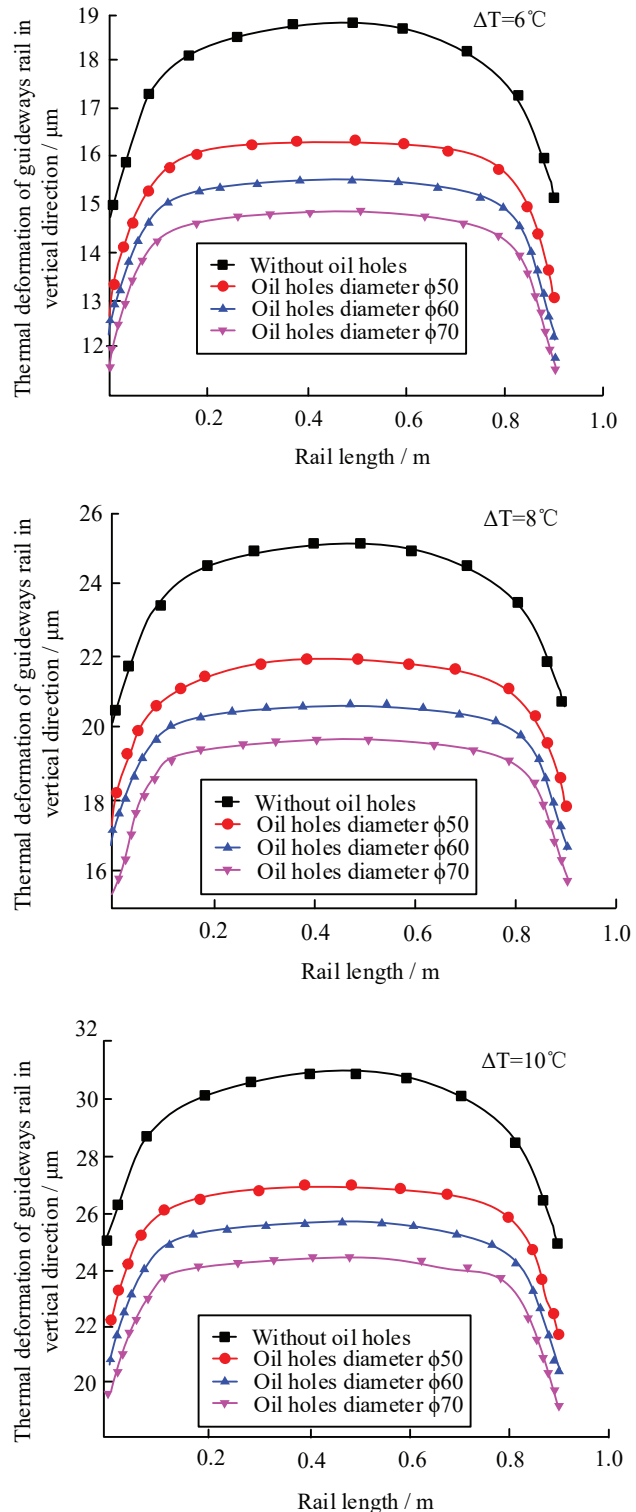
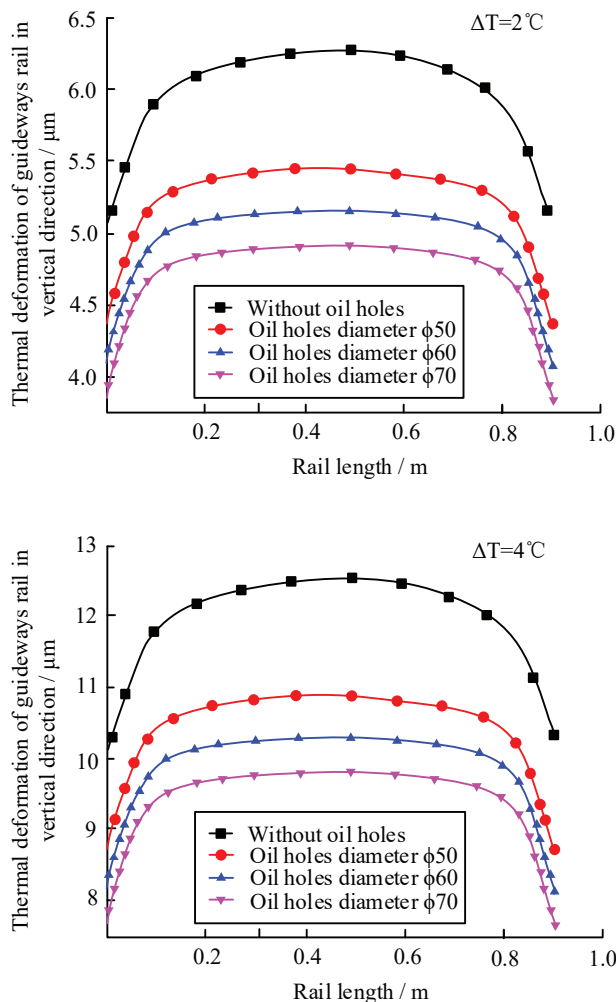


FIGURE6. INFLUENCE OF OIL HOLES ON THERMAL DEFORMATION OF GUIDWAYS RAIL IN VERTICAL DIRECTION

FIGURE 7. shows the form error of guideways rail. Under the same temperature rise, and along with the increase of oil hole diameter, form error is decreased, it can be reduced by a half, especially compared with guideways foundation without oil holes. Under the different temperature rise, the effect trend of oil hole diameter on form error of guideways rail shows consistency.

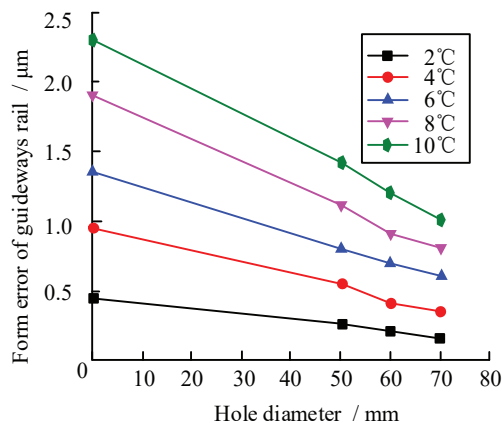


FIGURE7. INFLUENCE OF OIL HOLE DIAMETER ON FORM ERROR of GUIDEWAYS RAIL IN VERTICAL DIRECTION

CONCLUSION

In this paper a new structure of closed oil hydrostatic guideways is proposed, oil holes are set on foundation to improve the thermal stability by realizing the equalization of the distribution of temperature. Setting up the temperature mode of closed oil hydrostatic guideways foundation, the calculation of finite element method (FEM) indicates the new structure of closed oil hydrostatic is effective in reducing the thermal deformation and form error caused by uneven distribution of temperature,

(1) The simulation results show along with the increase of the oil hole diameter, the foundation had twice higher homogenizing temperature than that is without oil holes. the influences of oil holes on temperature distribution of guideways foundation, thermal

deformation of guideways rail and form error of guideways rail follow the same trend, the more homogeneous of temperature distribution, the smaller thermal deformation and form error of guideways rail.

(2) Along with the increase of oil hole diameter, maximum temperature difference of guideways foundation with oil holes is decreased, it can be reduced more than 50%, especially compared to guideways foundation without oil holes.

(3) Along with the increase of oil hole diameter, thermal deformation of guideways rail is decreased, it can be reduced about 30%, especially compared with guideways foundation without oil holes. form error of guideways rail is also decreased, it can be reduced nearly 50%, especially compared with guideways foundation without oil holes.

NOMENCLATURE

h	the convective heat transfer coefficient
Nu	Nusselt number
k	thermal conductivity of the fluid
l	characteristic length
T_i	initial temperature
ΔT	temperature rise
f	the homogenization of temperature distribution

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REFERENCES

- [1] McClure E R. The significance of thermal effects in manufacturing and metrology. Ann. CIRP, 1967; 15.
- [2] Bryan J B. International status of thermal error research. Annals of CIRP 1990; 39(2): 645-656.
- [3] Week M, Mckeown P, Bonse R, Herbst U.

Reduction and compensation of thermal error in machine tools. *Annals of CIRP* 1995; 44(2): 589-598.

[4] Aronson R B. War against thermal expansion. *Journal of Management in Engineering*. 1996; 116(6): 45-50.

[5] Jedrzejewski, J. Directions for improving the thermal stability of machine tools. In *Thermal Aspects in Manufacturing—Winter Annual Meeting ASME*. 1998; 30: 165-182 (ASME Production Engineering Division) .

[6] Nakamura, S, Kakino, Y. A Performance Evaluation of Preload Switching Spindle. *Journal of Japan Society of Precision Engineering*. 1994; 60(5): 688-692.

[7] Jedrzejewski J. Effect of Thermal Contact Resistance on Thermal Behaviour of the Spindle Radial Bearing, *International Journal of Machine Tools&Manufacture*. 1988; 28(4): 409-416.

[8] Nakamura S, Kakino Y. An Analysis and a Performance Evaluation of the Under-Race Lubrication Spindle at High Speed Rotation. *Journal of Japan Society of Precision Engineering*. 1994; 60(10): 1485-1489.

[9] Jedrzejewski, J. Machine Tool Designing with Thermal Criteria Taken Into Account. *Mechanik* 9. 1984; 460-466.

[10] Judd R L, Aftab K and Elbestawi M A An. investigation of the use of heat pipes for machine tool spindle bearing cooling. *Int. J. Mach. Tools Mf*, 1994; 34(7):1031–1043.

PARALLEL WIRES DRIVEN MICRO MANIPULATOR

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INSTRUCTIONS

In recent years, there are many kinds of micro positioners with such a linear stepping motor and high precision ball screw. Here we are proposing the parallel wires driven micro manipulator so that the micro tool and sensor device can be scanned on the target over the wider range with higher accuracy as well as high expandability. In our device, the tool holder is supported by two parallel wires that are connected with the pulleys. These pulleys can be rotated by the stepping motors with the reduction gears. The tool holder has the slider mechanism that is also connected with the wire from the upper frame so that the weight of the tool can be cancelled with the position feedback property. Thus the tool can be transported in the horizontal X axis and then the tool can be wound up and down at any position precisely.

The principle of this mechanism is described and several performance are discussed with the experimental setup.

PRINCIPLE OF PARALLEL WIRE DRIVEN MICRO MANIPULATOR

In order to determine the spatial position of the tool with high accuracy over the wider range, it's well-known techniques to use the parallel wires. In Fig.1, it is illustrated that the principle of our parallel wires driven positioner is composed two pairs of parallel wires that are connected to the pulleys at the both side. In the system, the upper side pulley at each side are set on the Z axis linear stage so that this pulley can be move upward and downward. This layout can allow to control the position and the angle of the target tool without any geometrical restriction.

Furthermore, the micro tool in the tool holder can be positioned with the wire that is wound up and down by another pulley set on the upper frame. Therefore, the micro tool in the tool holder can be transported and positioned as well as the tool can be easily penetrated into the target. Our final goal is to apply this system for such medical surgery with low electromagnetic noise.

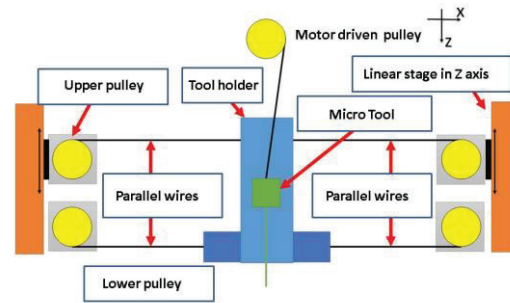


FIGURE 1. Principle of parallel wires driven micro manipulator

When the angle of the tool holder θ , the horizontal positions (x_1, y_1) at the point A of the tool holder and the point B (x_2, y_2) in Fig.2 can be represented as mentioned below.

$$x_1 = -a \sin \theta - \frac{b}{2} \cos \theta \quad (1)$$

$$y_1 = a \cos \theta - \frac{b}{2} \sin \theta \quad (2)$$

$$x_2 = -a \sin \theta + \frac{b}{2} \cos \theta \quad (3)$$

$$y_2 = a \cos \theta + \frac{b}{2} \sin \theta \quad (4)$$

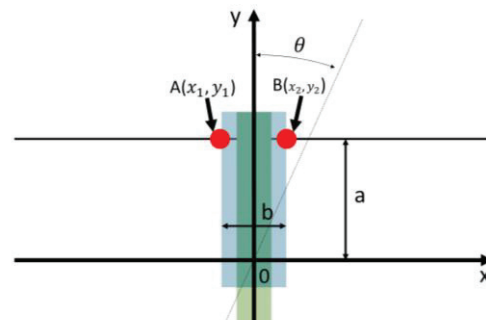


FIGURE 2. Coordination of the tool holder connected with the parallel wires.

The rotation of each pulley and the vertical position can be controlled precisely with the stepping motor and the linear motor so that both the angle and the position of the tool holder can be fixed under the geometrical relation.

EXPERIMENTAL SETUP

Figure. 3 shows the experimental setup of the parallel wires driven micro manipulator. Here an automatic positioning stage in Z axis can be controlled to adjust the position of the upper pulleys so that the geometrical relation can be satisfied at any inclined angle of the tool.

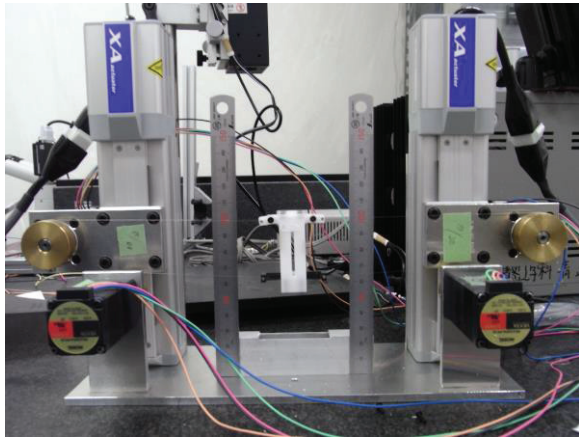


FIGURE 3. Experimental setup of parallel wires driven micro manipulator

EXPERIMENTAL RESULTS

As the primary experiments, the positioning resolution has been checked over the working range of 60mm. As shown in Fig.4, it is found that the minimum one step that is measured by the vision image tracking instrument is 7.5 micrometer with the standard deviation σ of 0.61 μ m. In Fig.5, another results of the step of 30 μ m is shown under the open loop control. It is also found that the good repeatability has been achieved with the deviation of 0.84 μ m over 30mm range. And the performance of the angle positioning has been also checked. The response with 0.1 degree/step is shown in Fig.6 while 10 degree/step is done in Fig.7. This results can indicate that this simple parallel wires manipulator has the long working range with fine positioning performance although the mechanical stiffness is not appropriate for such machine tooling.

In order to maintain the tool position at the specified position over the horizontal tool holder

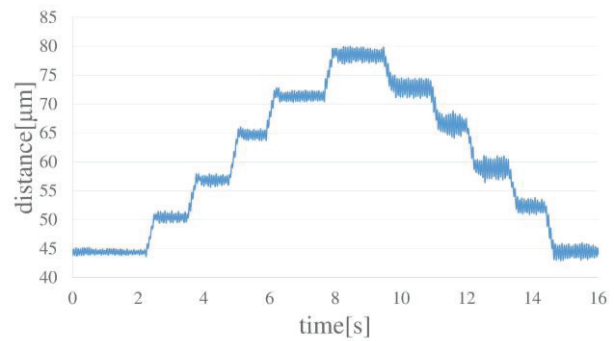


FIGURE 4. Positioning performance of 7.5 μ m/step with parallel wires driven micro manipulator

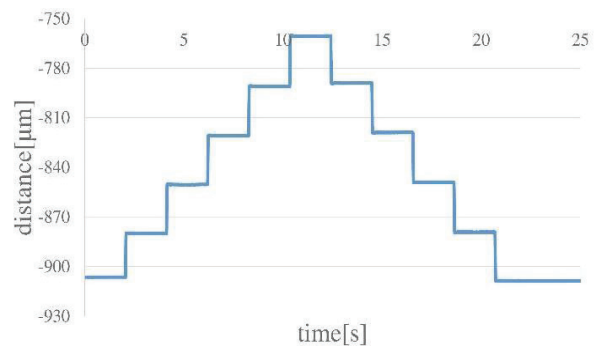


FIGURE 5. Positioning performance of 30 μ m/step with parallel wires driven micro manipulator

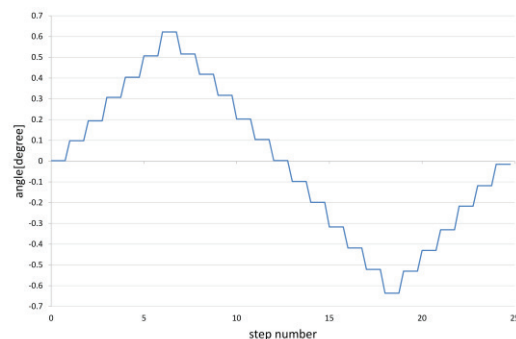


FIGURE 6. Angular positioning performance of 0.1 degree/step with parallel wires driven micro manipulator

travel, the tool is supported by another wire that is driven by the pulley as shown in Fig.8.

The tool position can be detected by a Hall position sensor that is embedded in the tool holder so that the tool position can be controlled by the pulley rotation over the horizontal moving range. The test tool for probing the target is employed in the tool holder as shown in Fig.10. The simple stainless steel needle is implemented into the tool holder. All of the positioning processes are managed by PC based controller. Typical tracking performance is shown in Fig.11. Here it is found that the tool can be positioned with this simple wire mechanism.

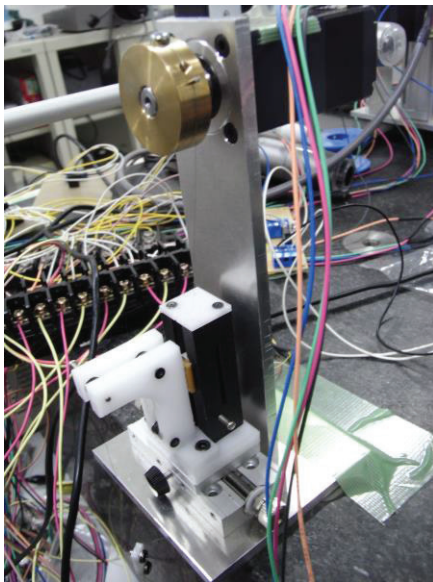


FIGURE 8 . Another pulley and wire for counterweight to maintain the tool position in Z-axis.



FIGURE 10 . Test tool for probing the target in the tool holder

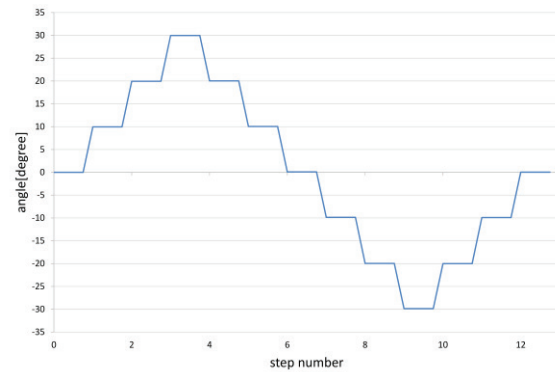


FIGURE 7 .Angular positioning performance of 0.1degree/step with parallel wires driven micro manipulator

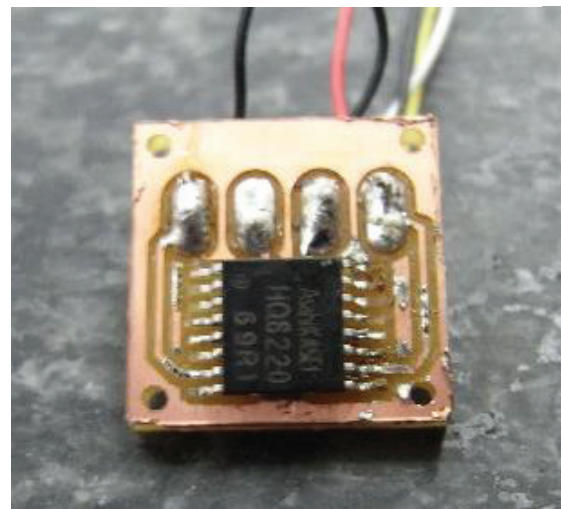


FIGURE 9 . Hall position sensor that can detect the vertical displacement and provide the feedback signal.

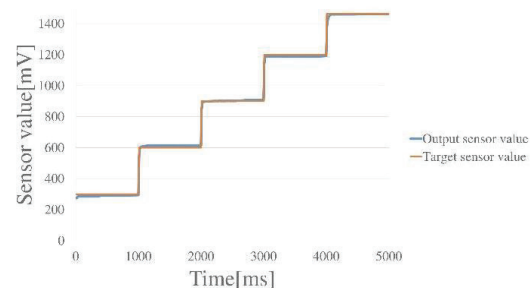


FIGURE 11 . Typical tracking performance of the wire positioner for tool.

As the typical application, the contact probe can be controlled with its position and angle so that it can check the circuit test around IC chip as shown in Fig.12.

CONCLUSIONS

With the help of parallel wires driven mechanism, the micro tool can be transported in the horizontal axis with high accuracy over 60mm travel and with the angular capacity of 60 degree. And the micro tool such the contact probe has been positioned precisely to check the electric circuit around the chip. In the future works, it is on going development to improve the system to care of such medical surgery application.

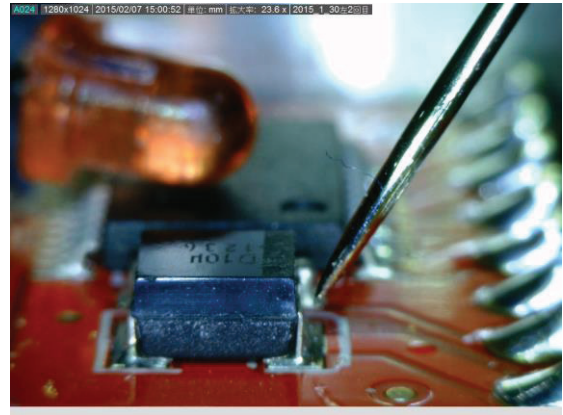


FIGURE 12 Application for the probing test for IC chip connection with the signal check detector.

MEASUREMENT OF SUB-MICROMETER FEATURES IN BOROSILICATE GLASS MICROPIPETTES

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ABSTRACT

Many commonly-used biological techniques make use of hollow glass needles called micropipettes. Micropipettes are typically fabricated by the end user, using a “pipette puller” that uses open-loop control to heat a hollow glass capillary while applying an axial force; however, variability between manufactured pipettes required a highly trained operator to qualitatively inspect each pipette as it is pulled using a microscope to ensure that it has the correct tip shape (e.g., outer diameter and cone angle). We have developed methods for automatically measuring micropipette tip outer diameter and cone angle from images taken using a microscope with a 100x objective with throughput times of < 30 seconds per pipette. Tip outer diameter measured compared favorably with the gold standard of scanning electron microscopy, differing by $0.35 \pm 0.36 \mu\text{m}$. Methods are described for repeatably constraining a pipette, automatically finding its tip in the microscope field of view, focusing on the pipette tip, and extracting tip outer diameter and cone angle measurements from the acquired images by applying a series of image processing algorithms. While we implemented these techniques using a custom-built microscope, the methods used are general and could be implemented on any commercial automated microscope.

INTRODUCTION

Glass micropipettes are hollow, tapered needles with tip inner diameter ranging from 0.5 to 75 μm . They are a critical component of biological techniques including microinjection [1], cell manipulation for *in vitro* fertilization [2], and electrical recording of neural activity [3], each of which requires micropipette tips of different diameters and taper angles (or “cone angles”). Typical applications may require hundreds to thousands of pipettes (e.g., patch clamp electrophysiology may consume 100 pipettes per day [3]). Micropipettes are susceptible to breakage and dust contamination, and so they are

preferably fabricated immediately before use using commercially-available “pipette pullers” (e.g., Sutter Instrument, Novato, CA) [1]. Pipette pullers use open-loop control to heat a glass capillary while applying an axial load, using either a constant force [1] or constant velocity [4], until the capillary necks and separates into two virtually identical micropipettes.

Frequent (weekly to monthly) calibration is necessary to “tune” pipette pullers, in which an operator iteratively adjusts pulling parameters (e.g., heating time, cooling time, pull velocity) to achieve micropipettes of the desired geometry (e.g., tip diameter, cone angle). At each iteration, a highly trained operator pulls a micropipette, qualitatively inspects the tip geometry by viewing under a microscope, uses experience to decide how to adjust puller settings, and repeats. This process can take several hours and hundreds of iterations to obtain suitable micropipettes. Operators typically desire micropipettes within 20% of target tip diameter (e.g., nominal 1 μm o.d. pipettes for electrophysiology applications should be within 0.8-1.2 μm o.d. [personal communication]). Cone angle tolerances are highly application dependent.

Due to the significant time required for micropipette inspection during fabrication, a method for rapid, automated measurement of micropipette tip geometry is therefore desirable, with diameter and cone angle measurement resolution comparable to human capabilities. Such a method could also enable automatic closed-loop micropipette fabrication, in which pipettes are automatically and iteratively pulled and measured.

DESIGN AND FUNCTIONALITY

Microscope and micropipette fixture

We implemented the pipette inspection methods using a custom-built microscope (shown in Fig. 1a),

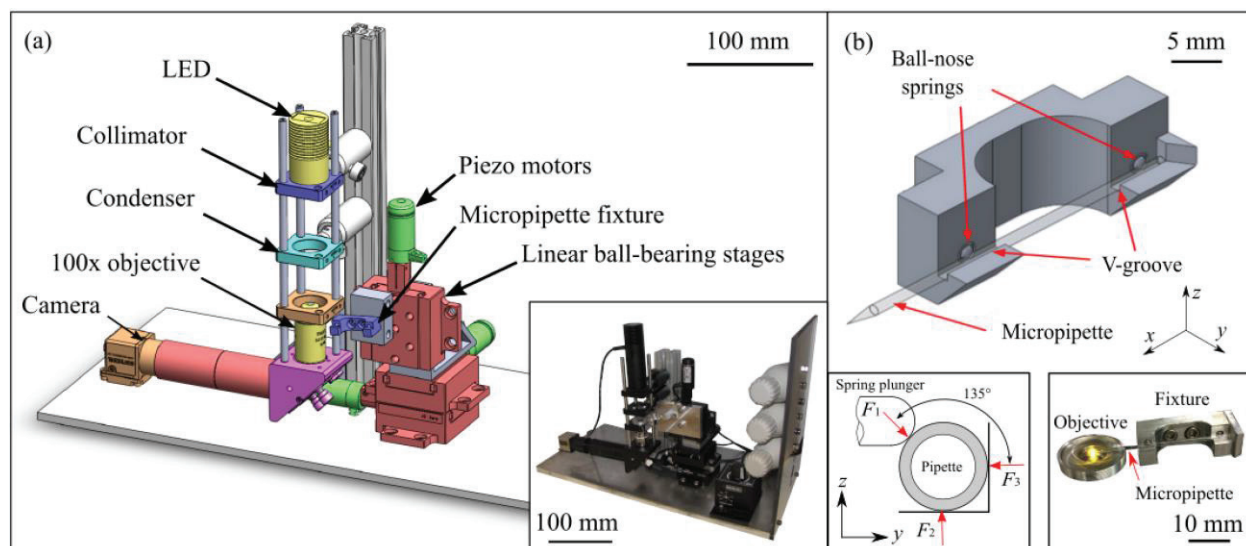


FIGURE 1. (a) Automated measurement of glass micropipette tip geometry was implemented on a custom-built microscope and motorized stage, but the methods described could be adapted to any commercial automated microscope. (b) Using a kinematic constraint mechanism, a pair of ball-nose spring plungers preload the pipette in aluminum V-grooves, resulting in intersecting lines of force (see left insert). Pipettes are constrained in yz by the V-grooves and registered to a hard stop in x . Photo (inset) shows fixture as fabricated, positioning a pipette over the objective.

but the same methods could be implemented on any commercial automated microscope. The optical system uses an LED (Thorlabs, 590 nm), collimator (Thorlabs), condenser (Thorlabs), objective (100x), and camera (Edmund Optics). The objective lens is immersed in 0.2 μm -filtered deionized water. A motorized three-axis positioner was assembled using three linear ball-bearing stages (Newport 460A) and three linear piezo motors (Newport PZA12). A custom aluminum fixture (Fig. 1b) accepts a micropipette in a V-groove and positions the tip in front of the objective. Ball-nose springs preload the pipette into the V-groove in the radial direction, and the pipette is preloaded against a hard stop in the axial direction. The micropipette fixture is secured to the three-axis positioner at a 15° angle to allow the tapered tip to be positioned within the working distance of the objective. The camera and piezo motors were controlled via an interface developed in MATLAB. A background image is captured when the system is initialized, and is subtracted from future frames to account for non-uniform illumination and any contamination of the optics.

Pipette tip-finding algorithm

To find the pipette tip in the field of view, a search strategy was developed that moves the pipette based on features visible in each image acquired by the camera, as shown in Fig. 2. For each frame, a boundary (30 pixels wide) is identified along each edge of the image. In each boundary region, if the maximum pixel intensity is greater than a

specified threshold, the edge is encoded as “1”, indicating that part of the pipette is visible along that edge of the image. Otherwise, no part of the pipette is visible along that edge of the image and the edge is encoded as “0”.

Fig. 2 shows the eight combinations of edge conditions encountered in normal operation and the pipette movements that result from each. When a pipette is first inserted and nothing is detected in the field of view, the pipette is moved in 10 μm steps in the $+x$ direction until another case is detected. The result of this search strategy is that the pipette follows a “zig-zag” path along the edge of the pipette until the tip is in the field of view.

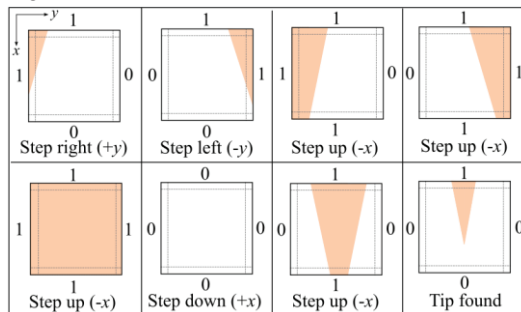


FIGURE 2. Eight edge conditions for the pipette tip search strategy. The pipette is moved in 10 μm steps depending on whether some part of it is (1) or is not (0) present in a boundary region along each edge of the image. The pipette is moved iteratively until the final condition (lower right) is encountered and the tip is in the field of view.

Autofocus algorithm

To focus on the pipette tip after it is in the field of view, we implemented a basic autofocus algorithm based on the Laplacian variance focus measure [5]. A flowchart of the autofocus algorithm is shown in Fig. 3. An initial image ($i = 1$) is acquired and its focus measure, F , is computed. The pipette is moved one step in the $+z$ direction (away from the objective), then another image is acquired. If F decreases on the first step, then the direction is reversed since the initial step was in the wrong direction. If F increases, the pipette continues stepping in the same direction until F decreases. When F decreases, a local maximum in focus has been crossed. Initial step direction is then reversed and the method is then repeated with the next step size (sequentially: $20\ \mu\text{m}$, $5\ \mu\text{m}$, $2\ \mu\text{m}$, $0.5\ \mu\text{m}$) to obtain increasingly fine focus on the pipette tip.

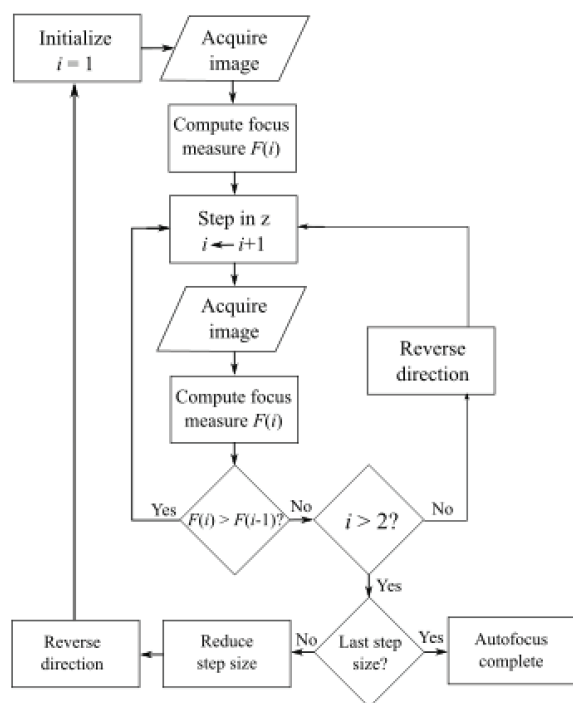


FIGURE 3. An autofocus algorithm captures pipette images, computes their Laplacian variance focus measure, then moves the pipette accordingly to maximize focus. The pipette is moved in one direction until the focus measure stops increasing, then the direction is reversed and the process is repeated with the next step size (sequentially: $20\ \mu\text{m}$, $5\ \mu\text{m}$, $2\ \mu\text{m}$, $0.5\ \mu\text{m}$) to obtain increasingly fine focus on the pipette tip.

Tip outer diameter measurement algorithm

An algorithm was developed to extract outer diameter measurements from a cropped image of the pipette tip. Fig. 4 illustrates the image processing steps on a representative pipette image. The original pipette image (Fig. 4a) is cropped (Fig. 4b). The cropped image is low-pass filtered by convolving with a discrete Gaussian kernel of standard deviation σ and kernel size n ($\sigma = 3\ \text{px}$, $n = 10\ \text{px}$) to remove high spatial frequency components and reduce effects of noise (Fig. 4c). Next, Canny edge detection [6] is applied to the filtered image to trace the inner and outer walls of the pipette, resulting in a binary array the size of the cropped image (Fig. 4d). A linear Hough transform [7] is applied to fit straight lines to the inner and outer walls of the pipette (Fig. 4e). The pipette centerline is then identified in the image as follows: the centerline is defined by a point (the centroid of the Canny edge detection array) and a slope (the average slope of the two outer lines identified using the Hough transform). To identify the axial position corresponding to the pipette tip, the pixel intensity along the pipette centerline is plotted (Fig. 4f), beginning from the top row of the image. The position of the tip along the centerline is identified as the location where the intensity along the centerline is halfway between the absolute maximum and the following local minimum. The tip outer diameter is then measured as the distance between the two outer Hough lines at the axial tip position identified from the centerline intensity plot.

Scanning electron microscopy

Scanning electron microscopy (SEM) is the “gold standard” for imaging micropipette tips. To evaluate the accuracy of our measurements, twenty-seven pipettes with tip o.d. ranging from $0.66\ \mu\text{m}$ to $47.8\ \mu\text{m}$ were imaged using our system and using a Hitachi SU8230 FE-SEM at 40,000x magnification. Samples were imaged head-on, with electron energies of 1.5 KeV or less at $1\ \mu\text{A}$ using the secondary electron detector. Outer diameter measurements were extracted from the acquired images using ImageJ by manually fitting an ellipse to the pipette circumference.

RESULTS

Repeatability of pipette tip positioning

A single micropipette tip can be positioned in the fixture with repeatability defined by $\sigma_x = 16\ \mu\text{m}$, $\sigma_y = 16\ \mu\text{m}$, $\sigma_z = 40\ \mu\text{m}$ (coordinate system shown in Figs. 1-2). However, because pipettes vary in length and tips are not axisymmetric, variability in tip positioning is dominated by geometric

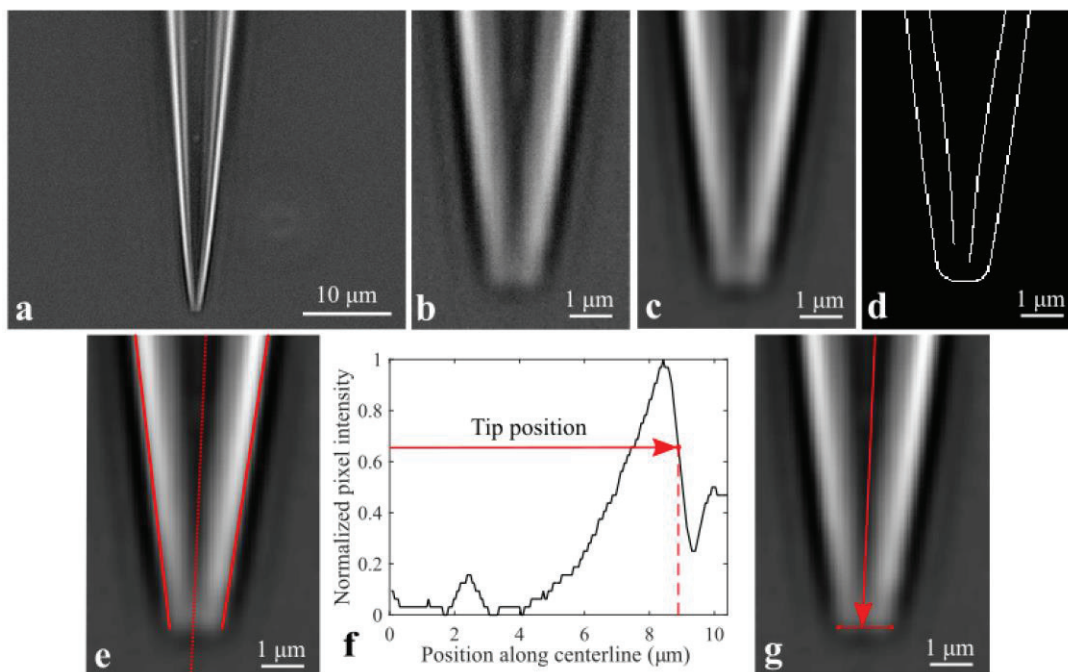


FIGURE 4. A series of image processing computations were applied to identify the tip outer diameter and cone angle. Pipette tip images: (a) original; (b) cropped; (c) low-pass filtered; (d) Canny edge detection; (e) Hough transform lines and pipette centerline. (f) Normalized intensity of pixels along the pipette centerline, with tip axial position labeled. (g) Image of pipette with tip position from (f) used to identify the cross section corresponding to the tip to measure outer diameter.

differences between pipettes. Ten different pipettes were pulled and inserted into the fixture, and their tip position variation was defined by $\sigma_x = 116 \mu\text{m}$, $\sigma_y = 41 \mu\text{m}$, $\sigma_z = 60 \mu\text{m}$, where x is along the axis of the pipette. σ_x is especially large because pipettes vary mainly in length. Further, the angle (about its axis) at which a pipette is inserted into the fixture affects its tip position; rotating the pipette about its axis causes the tip position to vary on the order of tens of microns. These repeatability measurements provide an estimate of the volume that must be “searched” to find the tip in the $29 \mu\text{m} \times 45 \mu\text{m}$ x-y field of view of the optical system when a new pipette is placed in the fixture.

Repeatability of measurements

Repeatability was assessed in two different ways to characterize the different sources of error between repeated measurements. In the first study, a single pipette was imaged, moved out of the field of view, moved back into the field of view, and imaged again. This process was repeated thirty times, resulting in thirty outer diameter and cone angle measurements on the same pipette. This allowed characterization of the sensitivity of

the image processing algorithms to small changes in the acquired image.

Measuring the same pipette thirty times as described resulted in outer diameter measurements of $2.89 \pm 0.08 \mu\text{m}$ and cone angle measurements of $27.5 \pm 1.13^\circ$ (mean $\pm$ SD). With 95% confidence, repeated outer diameter and cone angle measurements differed by less than and $0.16 \mu\text{m}$ and 2.2° respectively.

In the second study that was performed, a pipette was placed in the fixture and imaged, then it was removed, replaced in the fixture, and imaged again. This process was repeated for twenty five different pipettes, resulting in two outer diameter and cone angle measurements for each. This allowed characterization of the effect of imaging the pipette from different angles, since the pipette was randomly rotated about its axis before it was replaced in the fixture. Additionally, this test allowed us to examine a greater range of pipette sizes.

Fig. 5 plots the two outer diameter measurements taken for each pipette. Mean outer diameter

measurements ranged from $2.2\ \mu\text{m}$ to $6.2\ \mu\text{m}$. Repeated measurements differed by $-0.01 \pm 0.20\ \mu\text{m}$ (mean $\pm$ SD, first measurement minus second). With 95% confidence, the difference between two repeated measurements was less than $0.38\ \mu\text{m}$, assuming errors are normally distributed. The assumption of normality is reasonable given the observed distribution of errors between repeat measurements (histogram not shown). This test-retest error confidence interval is substantially larger than the single-pipette 95% confidence interval of $\pm 0.38\ \mu\text{m}$. The observed difference between test-retest outer diameter measurements can plausibly be explained in part by asymmetry of the pipette tip.

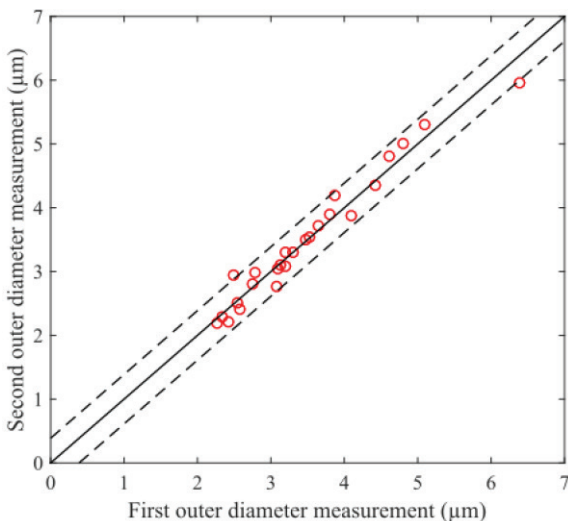


FIGURE 5. Results of repeated pipette outer diameter measurements on multiple pipettes. With 95% confidence, the first and second measurements differed by less than $0.38\ \mu\text{m}$ (dotted lines), assuming errors were normally distributed.

Fig. 6 plots the repeated cone angle measurements. The difference between repeated measurements was $0.38 \pm 2.78^\circ$. Unlike the diameter measurements, the distribution of the difference between repeated cone angle measurements appeared bimodal and symmetric (histogram not shown). Making the conservative approximation that the distribution is normal, repeated cone angle measurements differ by less than 5.45° with 95% confidence. This is substantially larger than the single-pipette confidence interval of 2.2° , suggesting that replacing pipettes in the fixture contributes some error to repeated measurements, more so than for diameter measurements. However, the cause of the bimodal distribution of test-retest variations remains unclear.

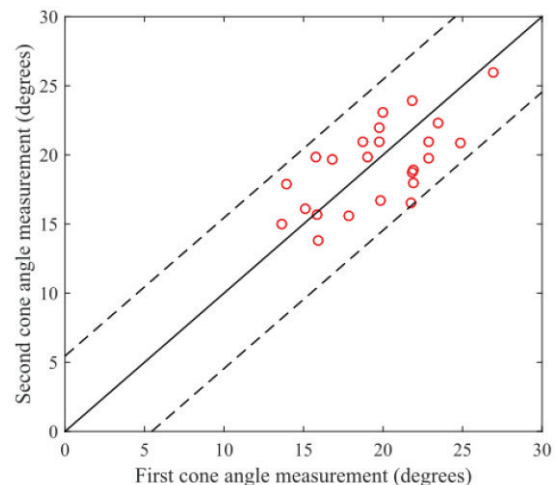


FIGURE 6. Results of repeated cone angle measurements on multiple pipettes. With 95% confidence, the first and second measurements differed by less than 5.45° (dotted lines), making the conservative approximation that errors were normally distributed.

Accuracy of outer diameter measurements

Twenty seven pipettes were imaged using our optical system and using a Hitachi SU8230 FE-SEM. To mitigate sample charging, samples were imaged with electron energies of 1.5 KeV or less at $1\ \mu\text{A}$. Fig. 7 shows a representative head-on image used to measure tip outer diameter, and Fig. 8 plots SEM and optical measurements of the same pipettes. Our optical measurements differed from SEM by $0.35 \pm 0.36\ \mu\text{m}$. Subtracting the systematic error of $0.35\ \mu\text{m}$ improved agreement between our measurements and the “true” value provided by SEM (Fig. 8 plots the uncorrected data).

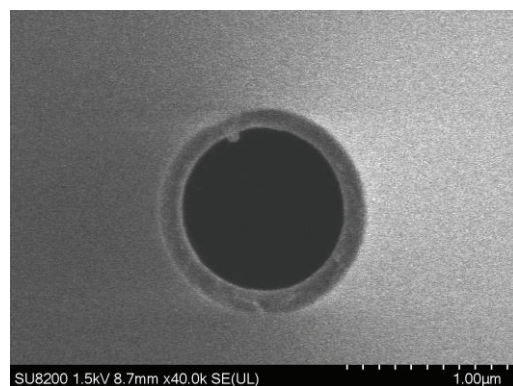


FIGURE 7. Representative head-on SEM image used to calculate the “true” micropipette tip outer diameter. The visible protrusion on the inner diameter is a filament annealed to the glass to

provide the capillary action needed to back-fill the pipette.

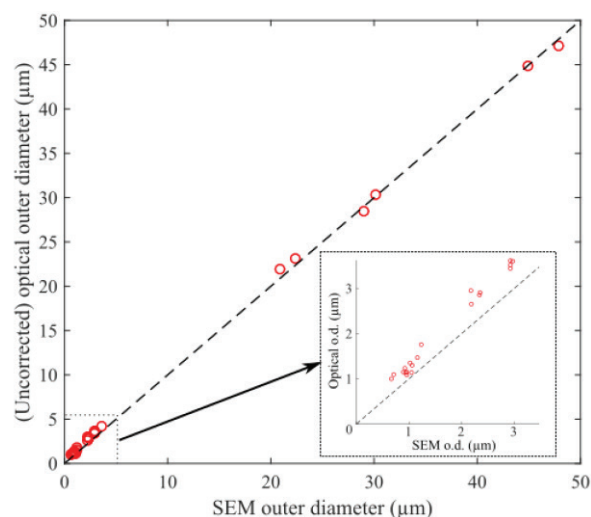


FIGURE 8. Comparison of our outer diameter measurements with the “gold standard” of SEM. Outer diameter measurements differed from SEM measurements by $0.35 \pm 0.36 \mu\text{m}$, before correcting for the measured systematic error. Inset: zoomed-in plot of uncorrected 1-3 μm o.d. pipette measurements.

CONCLUSIONS

Methods were developed for automated imaging and measurement of micropipette tips, including techniques for moving a pipette tip into the field of view of the optical system, focusing on the pipette tip, and extracting tip outer diameter and cone angle measurements from the acquired images. As implemented on a custom-built microscope, the methods achieve $\pm 0.38 \mu\text{m}$ repeatability in outer diameter measurement and 5.45° repeatability in cone angle measurement. Much of this variability between measurements appears to result from differences in which face of the asymmetric pipette tip is imaged, since repeated measurements taken of a single face of the same pipette were more consistent. For pipettes ranging from $0.66 \mu\text{m}$ to $47.8 \mu\text{m}$ in outer diameter, our outer diameter measurements differed from SEM by $0.35 \pm 0.36 \mu\text{m}$. Overall, performance was comparable to a trained human operator. We envision that with slight modifications, the algorithms described can be extended to inspect other commonly-used pipette tips, e.g. beveled. The methods we describe are useful both (1) on their own to reduce the time spent on micropipette inspection, and (2)

as a step toward a closed-loop, fully-automated micropipette fabrication and inspection system.

REFERENCES

- [1] Dean D, Gasiorowski J. Preparing injection pipettes on a Flaming/Brown pipette puller. Imaging: A Laboratory Manual. CSHL Press. New York: 2010.
- [2] Oesterle A. Pipette Cookbook 2015: P-97 & P-1000 Micropipette Pullers. Sutter Instrument: 2015.
- [3] Kodandaramaiah S, et al. Automated whole-cell patch clamp electrophysiology of neurons in vivo. Nature Methods. 2012; 9: 585-587.
- [4] Pak N, et al. An instrument for controlled, automated production of micrometer scale fused silica pipettes. Journal of Mechanical Design. 2011; 133.
- [5] Bueno-Ibarra M, et al. Fast autofocus algorithm for automated microscopes. Optical Engineering. 2005; 44.
- [6] Canny J. A computational approach to edge detection. IEEE Transactions on Pattern Analysis and Machine Intelligence. 1986; 6: 679-698.
- [7] Ballard J. Generalizing the Hough transform to detect arbitrary shapes. Pattern Recognition. 1981; 13: 111-122.

DESIGN, DEVELOPMENT AND TESTING OF PRECISION HYDROSTATIC ROTARY TABLE FOR MILLING-GRINDING MACHINE

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ABSTRACT

In milling-grinding machines, rotary tables with sub-micron accuracy are becoming more necessary. Usually, the radial error of rolling bearing is difficult to achieve the sub-micron accuracy. For this reason, the hydrostatic bearing with high rotation accuracy and large supporting stiffness were used in rotary table in milling-grinding machine. The present work describe the design, development and testing of precision hydrostatic rotary table consisting of a journal bearing and a double-thrust bearing. For precision design of components, radial and axial runout error models were established. Performed tests indicate that the preliminary measured accuracy highlight the proposed hydrostatic rotary table's potential for milling-grinding machine.

INTRODUCTION

Hydrostatic bearings have been used in many precision machine tools due to the high accuracy achieved by the error averaging effect of pressured oil film [1-5]. The motion error in hydrostatic bearing was mainly caused by variation of lubricating film clearance because of the existing structural parts geometric error. So, the internal relation between motion error and structural parts geometric error should be analyzed quantitatively for guiding the design and development of hydrostatic bearing. For motion error in hydrostatic bearing, Kane, et al [6] designed an angled surface self-compensating hydrostatic bearing and found error averaging ratio can reach 50 times, which indicated the rotary table with hydrostatic bearing may have higher accuracy than other supporting system, like YRT bearing.

DESIGN

The structural dimensions of hydrostatic rotary table were determined according to dimension of the workpiece to be machined (diameter of axis is 150 mm, rotating radius of rotary table is 150 mm), and the supporting way of the rotation axis was confirmed (hydrostatic double-

thrust bearing and radial bearing), as shown in Figure 1. The two thrust bearings connected with shaft with screws. The hydrostatic bearing has a constant pressure pattern. This is achieved by providing annular slit restrictor for each pad, and a separate temperature controlled hydraulic pressure station. A torque motor was employed to attach to the shaft directly. Velocity feedback and position feedback were realized by encoder which attached to the torque motor rotor. To avoid leakage, pressured dry oil was injected into the annular groove between the table and bearings.

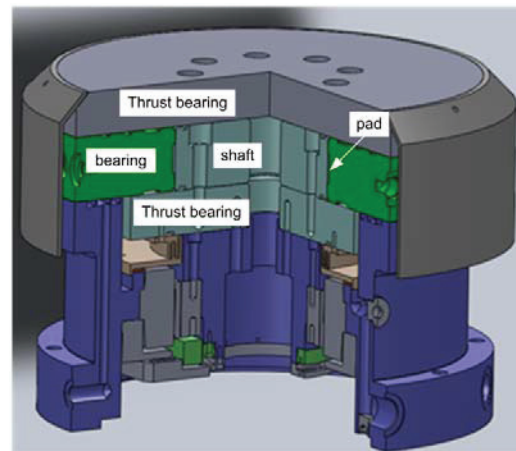


FIGURE 1. Hydrostatic rotary table design

MODEL AND PRECISION DESIGN

Error Modeling

Under error averaging effect of pressured oil film in hydrostatic bearing, radial runout error and axial runout error are always less than the structural parts geometric errors. Assuming that the shaft and bearing have elliptical errors in the hydrostatic journal bearing, and the hydrostatic double-thrust bearings have perpendicularity error and flatness errors. In order to quantify the relationship between rotary table accuracy and geometric errors, an error averaging model was established by using Reynolds

equation, pressure boundary conditions, flux continuity equation of the oil pad land and kinetic equation in hydrostatic bearing. Figure 2 shows the schematic explanation of radial runout error calculation method which expand the whole journal bearing for simulation. Figure 3 shows the radial runout error simulation results under different shaft or bearing elliptical error values.

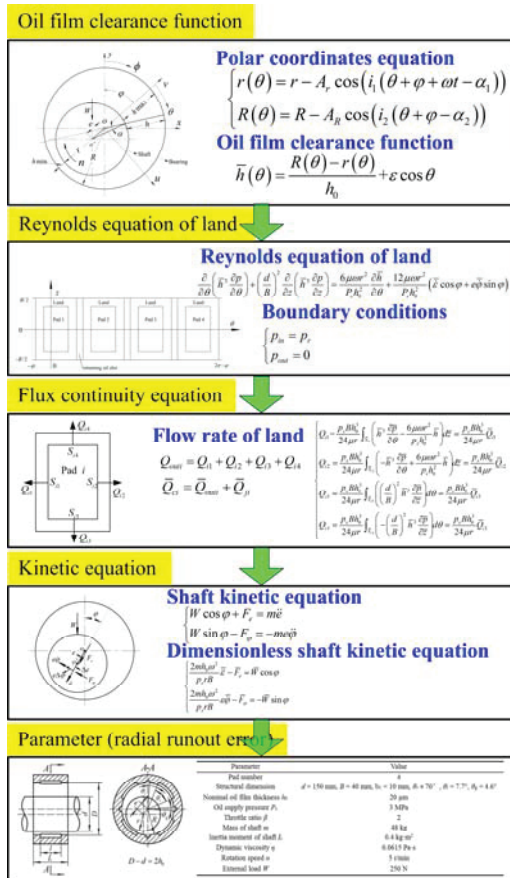


FIGURE 2. Hydrostatic journal bearing radial runout error model

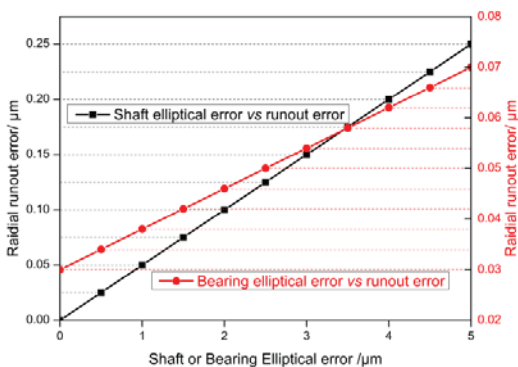


FIGURE 3. Relationship between shaft or bearing elliptical error value and radial runout error

Figure 4 shows the axial runout error calculation process considering perpendicularity error and flatness error in hydrostatic double-thrust bearing. Figure 5 shows the relationship between perpendicularity error value and axial run-out error. Figure 6 shows the relationship between flatness error of thrust bearing and axial run-out error.

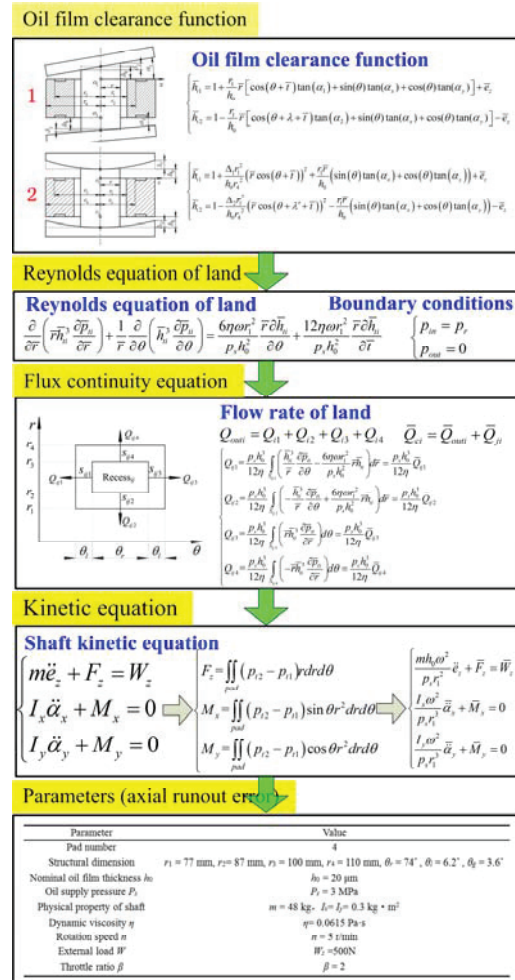


FIGURE 4. Hydrostatic thrust bearing axial runout error model

Precision Design

According to accuracy indexes of hydrostatic rotary table, both the radial runout error and axial runout error smaller than 0.2 μm . Considering the simulation results in Figure 3, 5 and 6, and the produced errors during the

assembly processes, for journal bearing, shaft roundness error is $2\text{ }\mu\text{m}$, bearing roundness error is $5\text{ }\mu\text{m}$. For thrust bearing, perpendicularity error is $5\text{ }\mu\text{m}$, thrust bearing flatness error is $5\text{ }\mu\text{m}$.

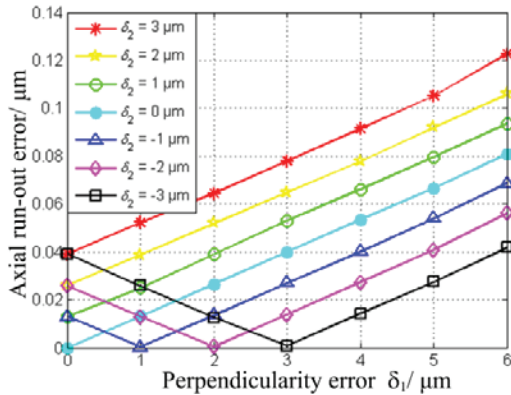


FIGURE 5. Relationship between perpendicularity error value and axial run-out error

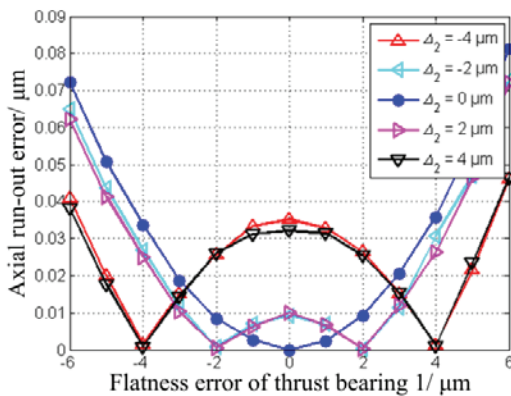


FIGURE 6. Relationship between two flatness errors and axial run-out error

DEVELOPMENT

The components for the rotary table were finish machined at a local CNC shop. Before assemble the components together, the roundness of shaft and bearing, and the flatness of two thrust bearings need to be closed measured. All the assembly processes were finished in a temperature controlled room. Firstly, we connected the shaft and one of the two thrust bearing by eight screws. And reduce the perpendicularity error between shaft axis and thrust bearing face as soon as possible by adjusting the screws preload. Secondly, inserted the shaft into the journal bearing carefully

because fit clearance ($40\text{ }\mu\text{m}$) is small, and connected another thrust bearing with it. Thirdly, assemble the annular slit restrictor to every pad, and inject the pressured oil into. The fit clearance of the hydrostatic double-thrust bearing cannot be measured directly because it is not accessible once the parts are assembled. So what need to be ensured before operation is the levitation height about $20\text{ }\mu\text{m}$ in the vertical direction for thrust bearing. After the aforementioned processes, turned the rotary table over and assemble the torque motor, the encoder and other auxiliary equipments. Figure 7 shows the developed hydrostatic rotary table.

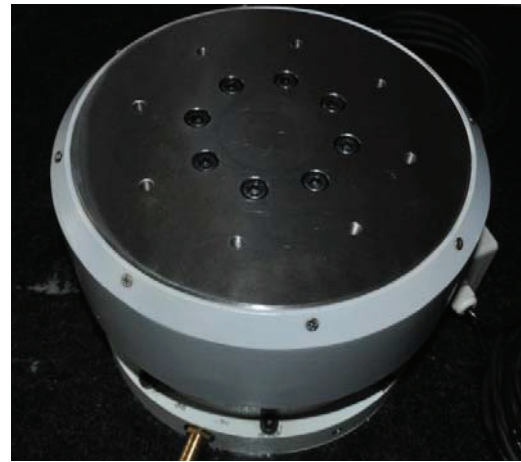


FIGURE 7. Image of hydrostatic rotary table

RESULTS

The measuring environmental conditions were as follows: room temperature was controlled within $21\pm 0.1^\circ\text{C}$, the humidity was 45.4%. The measurement setup was shown in Figure 8. Tesatronic TT80 Inductance micrometer and high precision inductive gauge head GT21 (repeatability $0.01\text{ }\mu\text{m}$) were used to measure the two runout errors. The standard sphere has a diameter 45mm. For data acquisition, sample interval is 200ms, 60 sampling points during every revolution, three revolutions were needed. Figure 9 and 10 are radial runout error and axial runout error measurement results, respectively. It's important to note that the eccentric value between the shaft axis and standard sphere axis was contained in the measured value. The runout error = Measured value – eccentric value.

According to the value in the precision design, the calculated radial and axial runout error were $0.06\text{ }\mu\text{m}$ and $0.064\text{ }\mu\text{m}$, respectively. The

comparison between simulation results and measurement results was shown in Table 1. The difference may be induced by the assembly errors or oil film pressure variation in the pads.

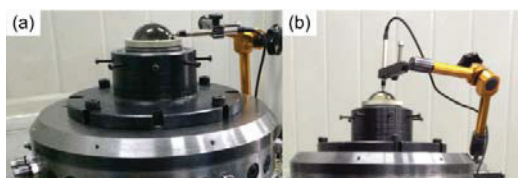


FIGURE 8. Measurement setup image (a) radial runout error (b) axial runout error

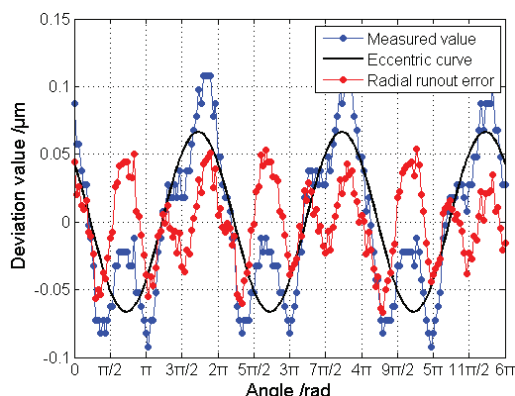


FIGURE 9. Measurement results of radial runout error

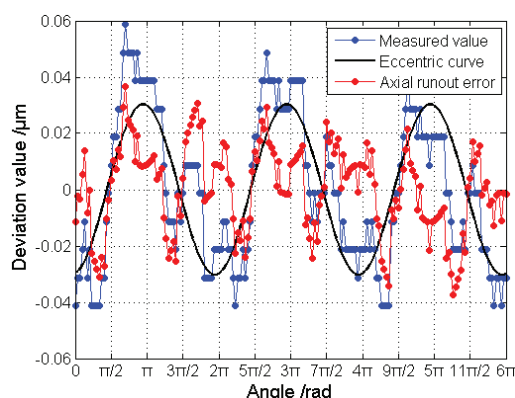


FIGURE 10. Measurement results of axial runout error

CONCLUSIONS

Accuracy tests performed on hydrostatic rotary table shows that the design and development has been successfully achieved. The radial and axial runout error model had been established based on error averaging effect by pressured oil film in hydrostatic bearings. Also the

measurement results verified the correctness of the precision design which derived from the two models. In the future, we will integrate the rotary table with milling-grinding machine and examine the machining accuracy.

TABLE 1. Comparison between simulation results and measurement results

Rotary table	Simulation result (μm)	Measurement result (μm)
Radial runout error	0.06	0.12
Axial runout error	0.064	0.07

ACKNOWLEDGEMENT

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REFERENCES

- [1] Noguchi S, Miyaguichu K. An evaluation method of radial accuracy for hydrostatic air spindles considering radial movement of the rotating center. *Precision Engineering*. 2003; 27: 395-400.
- [2] Wasson K.L. A comparison of rolling element and hydrostatic bearing spindles for precision machine tool applications. *Proceedings ASPE, Precision Bearings and Spindles*, 2007.
- [3] Hwang J, Shim J.Y, Kim J.H, Park C.H. Error motion estimation of a hydrostatic spindle. *Proceedings of the 26th Annual Meeting of the American Society for Precision Engineering*, 2011.
- [4] Kashchenevsky L, Johnson D. Theoretical analysis of rotational accuracy for thrust hydrostatic bearings. *Proceedings of the 22th Annual Meeting of the American Society for Precision Engineering*, 2007.
- [5] Shim J, Hwang J, Park C.H. Simulation and experimental investigation on radial run-out error of rotary air bearing tables. *Proceedings of the 26th Annual Meeting of the American Society for Precision Engineering*, 2011.
- [6] Kane N.R, Sihler J, Slocum A.H. A hydrostatic rotary bearing with angled surface self-compensation. *Precision Engineering*. 2003; 27: 125-139.

MODELING OF LONGITUDINAL-BENDING COUPLED ELLIPTICAL VIBRATION CUTTING TOOL USING TIMOSHENKO BEAM THEORY

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INTRODUCTION

Elliptical vibration cutting was originally proposed to improve the cutting performance by adding two-dimensional vibrations to the cutting tool [1]. The intermittent nature of the process reduces the contact time between the cutting tool and workpiece, which makes it possible to cutting ferrous materials with a diamond tool [2]. The overlapping of elliptical trajectories significantly decreases the effective uncut chip thickness during cutting, which makes the process favorable in processing difficult-to-cut and brittle materials [3, 4]. The concept of elliptical vibration cutting was later expanded to other areas, such as surface texturing [5-7] and imprinting [8].

The key component in the elliptical vibration cutting and other inspired processes is a vibration generator that could deliver elliptical vibrations at the tool tip at an ultra-high frequency (usually in the ultrasonic range). The resonant vibrations of the structure is often adopted in order to achieve a large vibration amplitude at an ultrasonic frequency. The two-dimensional vibrations are achieved by utilizing the coupled natural modes. Typical examples include coupling of two bending vibrations of a beam [1, 2], longitudinal-bending coupled vibrations of a beam [4, 5], coupling of a complex structure [9]. There, however, lacks a theoretical understanding of the tool structure dynamics and a systematic way for optimal design.

To address the above-mentioned problems, this paper presents an analytical modeling approach to describe the system dynamics of a longitudinal-bending coupled vibration tool. The design principle is to match the resonant frequencies of the longitudinal and bending vibrations of a beam, so that the two orthogonal vibrations will be simultaneously excited at a single working frequency. The tool tip will follow an elliptical vibration trajectory due to the coupled vibrations in the transverse and longitudinal directions. In this paper, the transverse (bending) and longitudinal vibration mode shapes and

modal frequencies are analytically described and solved. The design is optimized by minimizing the difference between the longitudinal and transverse modal frequencies. The proposed model is also useful for describing the tool structure dynamics when studying machine-tool dynamics in elliptical vibration cutting.

MODELING

To generalize the problem, the tool structure is modeled as a straight beam, which is composed of n segments and m simple supports as shown in FIGURE 1. The whole beam cross-section area is non-uniform, as is often adopted in the stepped horn design; while each segment is assumed to be a uniform and Timoshenko beam with height h_i , width b_i and length L_i . There are m simple supports. The support locations are always at the intersection between two segments. If two segments have the same cross-section area, as the case for S_1 , it can represent the general case when the support is located at the middle of a beam. The left end of the whole beam is assumed to be under the fixed condition; while the right end of the beam is assumed be a free end.

The simple support condition models the fixture in the actual tool design; however, it over-constrain the model. To more realistically model the fixture structure, two springs in the transverse and axial directions are used to replace the simple support condition, as shown in FIGURE 2. The two spring constants can be set to very large numbers to represent the simple support condition. Alternatively, they can be set to

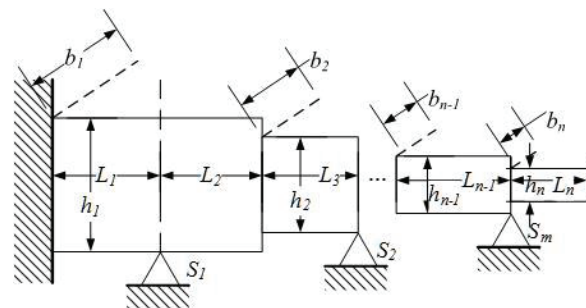


FIGURE 1. Generalized model of the tool.

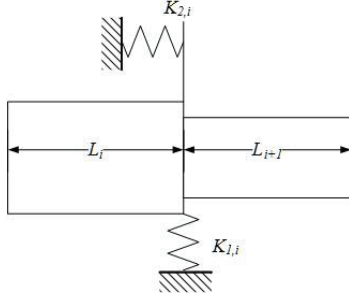


FIGURE 2. Model of the support fixture.

different values to model the anisotropic stiffness of the fixture.

The longitudinal and transverse vibrations of the beam are assumed to be independent. For transverse vibrations, the beam is modeled according to Timoshenko's theory. The bending motion can be described by the transverse displacement W and the rotation angle Φ . The differential equation for free vibrations of the i th segment can be derived using the separable solutions to be [10],

$$\frac{d^4 W_i}{dx^4} + (\sigma + \tau) \frac{d^2 W_i}{dx^2} + (\sigma\tau - \alpha_i) W_i = 0 \quad (1)$$

where $\tau = \frac{\rho\omega^2}{kG}$, $\sigma = \frac{\rho\omega^2}{E}$, and $\alpha_i = \frac{A_i\rho\omega^2}{EI_i}$.

E is the elastic modulus; G is the shear modulus; $I_i = \frac{1}{12} b_i h_i^3$ is the area moment of inertia of the i th segment for a rectangular cross-section; ρ is the material density; A_i is the cross-section area for the i th segment; κ is the Timoshenko shear coefficient.

The general solutions of the transverse displacement and the rotation angle take the form as,

$$W_i(x) = C_{1,i} \cosh \lambda_{1,i} x + C_{2,i} \sinh \lambda_{1,i} x + C_{3,i} \cos \lambda_{2,i} x + C_{4,i} \sin \lambda_{2,i} x \quad (2)$$

$$\Phi_i(x) = C_{2,i} q_{1,i} \cosh \lambda_{1,i} x + C_{1,i} q_{1,i} \sinh \lambda_{1,i} x - C_{4,i} q_{2,i} \cos \lambda_{2,i} x + C_{3,i} q_{2,i} \sin \lambda_{2,i} x \quad (3)$$

where

$$\lambda_{1,i} = \left(\sqrt{\left(\frac{\sigma - \tau}{2} \right)^2 + \alpha_i} - \frac{\sigma + \tau}{2} \right)^{\frac{1}{2}} \quad (4)$$

$$\lambda_{2,i} = \left(\sqrt{\left(\frac{\sigma - \tau}{2} \right)^2 + \alpha_i} + \frac{\sigma + \tau}{2} \right)^{\frac{1}{2}} \quad (5)$$

$$\lambda_3 = \sqrt{\frac{\rho L^2 \omega^2}{kG}} \quad (6)$$

$$q_{1,i} = \frac{\lambda_3^2 + \lambda_{1,i}^2}{\lambda_{1,i}} \quad (7)$$

$$q_{2,i} = \frac{\lambda_3^2 - \lambda_{2,i}^2}{\lambda_{2,i}} \quad (8)$$

Transfer matrices are established to describe the relationship between the unknown coefficients in the general solutions for the consecutive segments. By combining the boundary conditions and the transfer matrices, one can solve for the eigen values of the system to get the system natural frequencies, ω . There are four compatible conditions to be satisfied between the beam segments, i.e., transverse displacement continuity, slope continuity, moment continuity and shear continuity. For the connection shown in FIGURE 2, the transfer matrix can be derived in Equation (9).

$$\begin{bmatrix} 1 & 0 & 1 & 0 \\ 0 & \lambda_{1,i+1} & 0 & \lambda_{2,i+1} \\ \frac{I_{i+1}}{I_i} \lambda_{1,i+1} q_{1,i+1} & 0 & \frac{I_{i+1}}{I_i} \lambda_{2,i+1} q_{2,i+1} & 0 \\ -\frac{K_{1,i}}{kG b_i h_i} & \frac{b_{i+1} h_{i+1}}{b_i h_i} (\lambda_{1,i+1} - q_{1,i+1}) & -\frac{K_{1,i}}{kG b_i h_i} & \frac{b_{i+1} h_{i+1}}{b_i h_i} (\lambda_{2,i+1} + q_{2,i+1}) \end{bmatrix} \begin{Bmatrix} C_{1,i+1} \\ C_{2,i+1} \\ C_{3,i+1} \\ C_{4,i+1} \end{Bmatrix} \quad (9)$$

$$= \begin{bmatrix} \cosh \lambda_{1,i} L_i & \sinh \lambda_{1,i} L_i & \cos \lambda_{2,i} L_i & \sin \lambda_{2,i} L_i \\ \lambda_{1,i} \sinh \lambda_{1,i} L_i & \lambda_{1,i} \cosh \lambda_{1,i} L_i & -\lambda_{2,i} \sin \lambda_{2,i} L_i & \lambda_{2,i} \cos \lambda_{2,i} L_i \\ q_{1,i} \lambda_{1,i} \cosh \lambda_{1,i} L_i & q_{1,i} \lambda_{1,i} \sinh \lambda_{1,i} L_i & q_{2,i} \lambda_{2,i} \cos \lambda_{2,i} L_i & q_{2,i} \lambda_{2,i} \sin \lambda_{2,i} L_i \\ (\lambda_{1,i} - q_{1,i}) \sinh \lambda_{1,i} L_i & (\lambda_{1,i} - q_{1,i}) \cosh \lambda_{1,i} L_i & -(\lambda_{2,i} + q_{2,i}) \sin \lambda_{2,i} L_i & (\lambda_{2,i} + q_{2,i}) \cos \lambda_{2,i} L_i \end{bmatrix} \begin{Bmatrix} C_{1,i} \\ C_{2,i} \\ C_{3,i} \\ C_{4,i} \end{Bmatrix} \quad (10)$$

When there is no support condition at the connection, K_1 and K_2 are set to zero in Equation (9). The transfer matrix defined in Equation (9) is a function only of natural frequency, ω .

By combining the boundary conditions at the left and right ends of the beam, and the transfer matrices, the expression can be derived as Equation (10), where B is the matrix describing the free end condition at the right end of the beam.

To solve Equation (10) for non-trivial solutions, it is equivalent to solve the following equation for the modal frequencies of the system.

$$\det \begin{bmatrix} r_{11} - r_{13} & r_{12} + \frac{q_{11}}{q_{21}} r_{14} \\ r_{21} - r_{23} & r_{22} + \frac{q_{11}}{q_{21}} r_{24} \end{bmatrix} = 0 \quad (11)$$

The coefficients in the shape function W_1 , then can be solved by plugging ω back to Equation (10). Through the transfer matrix defined in Equation (9), all the coefficients can be determined.

Through a similar analysis, the axial vibration mode shapes and modal frequencies can be solved. The differential equation for the axial vibration for the i th segment can be expressed as,

$$\frac{d^2 U_i(x)}{dx^2} + \gamma^2 U_i(x) = 0 \quad (12)$$

where $\gamma^2 = \frac{\rho \omega^2}{E}$.

The general solution to the axial vibration is,

$$U_i(x) = D_{1,i} \sin \gamma x + D_{2,i} \cos \gamma x \quad (13)$$

The compatible conditions between two segments are axial displacement continuity and axial force continuity, which can be expressed by,

$$\begin{bmatrix} \frac{A_{i+1}}{A_i} - \frac{K_{2,i}}{A_i E} & -\frac{K_{2,i}}{A_i E} \\ 0 & 1 \end{bmatrix} \begin{Bmatrix} D_{1,i+1} \\ D_{2,i+1} \end{Bmatrix} = \begin{bmatrix} \cos \gamma L_i & -\sin \gamma L_i \\ \sin \gamma L_i & \cos \gamma L_i \end{bmatrix} \begin{Bmatrix} D_{1,i} \\ D_{2,i} \end{Bmatrix} \quad (14)$$

The spring constant K_2 is set to zero when there is no support condition at the connection.

RESULTS AND DISCUSSIONS

Three-step Beam Design

The modal frequencies solved from the transverse and longitudinal vibration equations

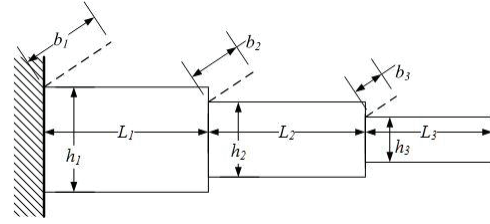


FIGURE 3. Three-step cantilever beam design.

are compared. The geometry of the structure can be varied to minimize the frequency difference between the transverse and longitudinal vibration modes. As an example and validation to the proposed model, a three-step cantilever beam, as shown in FIGURE 3, is considered. The input parameters to the model is summarized in TABLE 1. The cross-section areas are predefined for the model; while the lengths for three segments are variables to be optimized to minimize the frequency difference. The optimized lengths are determined to be $L_1 = 86.8 \text{ mm}$, $L_2 = 44.3 \text{ mm}$, and $L_3 = 29.3 \text{ mm}$. The calculated first 8 modal frequencies are compared with the FEM simulation results in TABLE 2. The mode shapes are identified as the bending (transverse) mode or the longitudinal mode. The corresponding mode shapes are normalized and plotted in FIGURE 4.

TABLE 1. Model input parameters.

Dimension	$h_1=30\text{mm}; h_2=25\text{mm};$ $h_3=10\text{mm}$ $b_1 = b_2 = b_3 = 20 \text{ mm}$ $60\text{mm} < L_1 < 100\text{mm}$ $20\text{mm} < L_2 < 60\text{mm}$ $10\text{mm} < L_3 < 30\text{mm}$
Material properties	$\rho = 7700 \text{ kg/m}^3$ $G = 79 \text{ GPa}$ $E = 210 \text{ GPa}$ $\kappa = \frac{10(1+\nu)}{12+11\nu} = 0.682$

TABLE 2. Modal frequencies calculated from the proposed model compared with the simulation results.

Mode description	Model results (Hz)	FEM results (Hz)	Error
1st bending mode	1,257	1,261	0.3%
2nd bending mode	5,543	5,373	3.2%
1st longitudinal mode	9,613	9,620	0.1%
3rd bending mode	10,701	10,037	6.6%
4th bending mode	17,144	16,916	1.3%
5th bending mode	26,376	26,817	1.6%
2nd longitudinal mode	26,378	25,897	1.9%
6th bending mode	36,362	36,347	0.04%

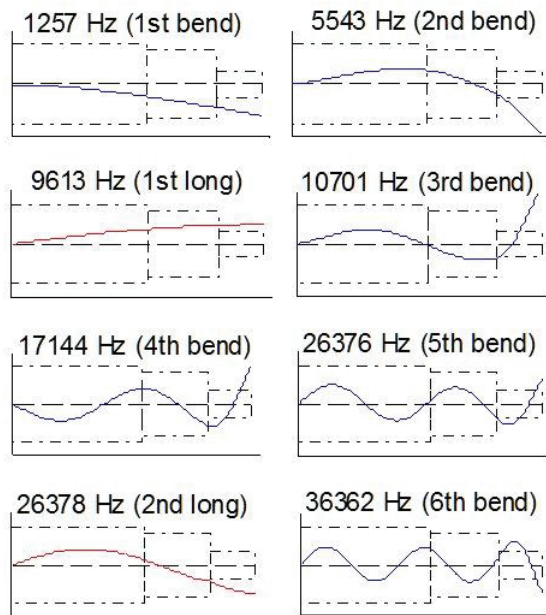


FIGURE 4. Mode shape and modal frequency.

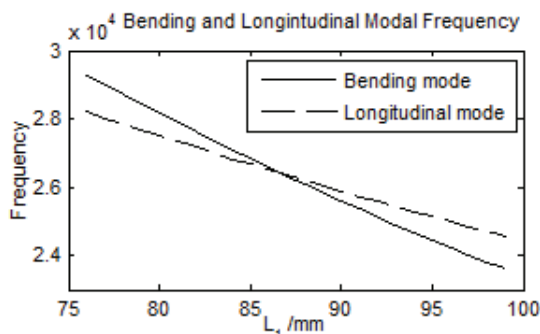


FIGURE 7. Influence of the beam length on the modal frequency.

Discussions on the optimal design

As shown in FIGURE 4, the modal frequencies of the 2nd longitudinal and the 5th bending modes are matched to be around 26.38 kHz. This frequency is then the working frequency for the tool, which will simultaneously excite the bending and longitudinal vibrations at the tool tip.

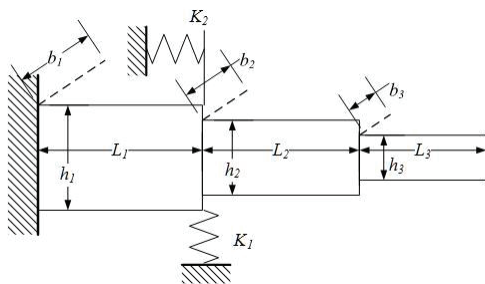


FIGURE 6. Three-step design with the support condition.

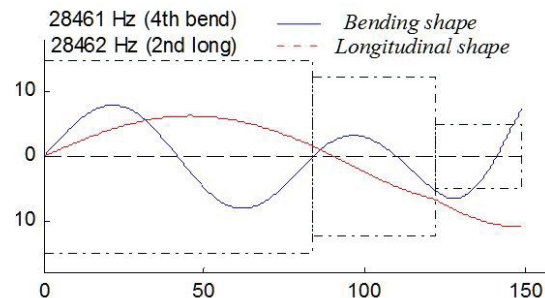


FIGURE 5. Coupled vibrations of the longitudinal and bending modes

When L_2 and L_3 are further fixed, the dependence of modal frequency on L_1 is plotted in FIGURE 5. With the increase in L_1 , the modal frequencies of the two orthogonal modes decrease, but with a different slope. There is an optimal point, where two frequencies are identical. By introducing additional variables of L_2 and L_3 , there are non-unique solutions for the minimal frequency difference. Additional design targets can be set, e.g. the resonant working frequency.

Discussions on the support condition

When there are additional support conditions, the resonant frequencies of different modes will change accordingly. As an example, an additional support condition in the three-step beam design is added as shown in FIGURE 6. The spring constants, K_1 and K_2 , are set to be 1×10^{12} N/m and 1×10^7 N/m respectively. The new optimal design is determined to have the dimensions: $L_1=84.0\text{mm}$, $L_2=38.3\text{mm}$, and $L_3=26.8\text{mm}$. The modal frequencies of the 4th bending and 2nd longitudinal modes are matched. The mode shapes of the coupled longitudinal and bending vibrations are plotted in FIGURE 7. It could be noted that due to the high spring constant in the transverse direction, the transverse motion is restricted to zero; while in the longitudinal direction, the relatively small spring constant does not force the longitudinal vibrations to zero.

CONCLUSIONS

In this paper, an analytical model is presented to describe the modal dynamics of a longitudinal-bending coupled elliptical vibration tool. The tool structure is modeled as a thick beam, or Timoshenko beam, which considers both the longitudinal and transverse vibrations. The modal frequencies and mode shapes are calculated and plotted. The optimal design is obtained by minimizing the frequency difference between the two orthogonal modes.

REFERENCES

- [1] Moriwaki T, Shamoto E. Ultrasonic elliptical vibration cutting, CIRP Annals-Manufacturing Technology. 1995; 44(1): 31-34.
- [2] Shamoto E, Moriwaki T. Ultraprecision diamond cutting of hardened steel by applying elliptical vibration cutting, CIRP Annals-Manufacturing Technology. 1999; 48(1): 441-444.
- [3] Nath C, Rahman M, Neo K S. Modeling of the effect of machining parameters on maximum thickness of cut in ultrasonic elliptical vibration cutting, Journal of manufacturing science and engineering. 2011; 133(1): 011007.
- [4] Suzuki N, et al. Elliptical vibration cutting of tungsten alloy molds for optical glass parts, CIRP Annals-Manufacturing Technology. 2007; 56(1): 127-130.
- [5] Suzuki N, Yokoi H, Shamoto E. Micro/nano sculpturing of hardened steel by controlling vibration amplitude in elliptical vibration cutting, Precision Engineering. 2011; 35(1): 44-50.
- [6] Guo P, Ehmann K F. An analysis of the surface generation mechanics of the elliptical vibration texturing process, International Journal of Machine Tools and Manufacture. 2013; 64: 85-95.
- [7] Guo P, et al. Generation of hierarchical micro-structures for anisotropic wetting by elliptical vibration cutting, CIRP Annals-Manufacturing Technology. 2014; 63(1): 553-556.
- [8] Zdanowicz E, et al. Nanostructure fabrication on germanium and silicon by nanocoining imprint technique, Precision Engineering. 2013; 37(4): 871-879.
- [9] Guo P, Ehmann K F. Development of a tertiary motion generator for elliptical vibration texturing, Precision Engineering. 2013; 37(2): 364-371.
- [10] Lin H-P, Chang S. Free vibration analysis of multi-span beams with intermediate flexible constraints, Journal of sound and vibration. 2005; 281(1): 155-169.

Microscopic-based imaging for precision measurement of micro scale flows around dynamically oscillating objects

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INTRODUCTION

The objective of this article is to detail the design and performance of a microscope-based imaging facility that comprises a piezoelectric actuator for oscillating one or more cylindrical rods immersed in a fluid containing small particles of diameter ranging from 10 - 150 μm . Within the exposure (or shutter) time of each camera frame it is possible to utilize multiple pulses with specified time intervals to identify the trajectory of individual moving particles. Images are presented of particle flows during which from 2 up to 20 pulses per frame are used to illuminate particles trajectories in a flow. This article contains detailed descriptions of the configuration of the current experimental set-up and instrument design, experimental procedure, and the preliminary results of particle velocity measurements. Several applications of this research including the sorting of particles in fluids based on their geometry, size and/or density, separating cells in biological samples based on their cell type [1], or the study of abrasive particle circulation vortex machining process for micro-polishing applications [2-5].

EXPERIMENTAL APPARATUS

Figure 1 shows a schematic representation of the apparatus of this development. The fluid is placed onto an adjustable platform and a manual micrometer is used to set the height of the oscillating fiber relative to the bottom of the transparent container. Adjustment of the platform enables the height of the fluid container to be positioned so that the tip of the rod nearly touches the bottom of the fluid container (contact being detected when the oscillating tip amplitude reduces). For imaging the induced streaming flows, it is important that the particles be near neutral density and sufficiently small to follow the flow of the fluid. Because of the small depth of field of the imaging system, two lasers are attached to lockable Heim Joint mounts that have

been set so that the two beams coincide at the focal plane of the lens, thereby providing visual feedback during camera adjustment.

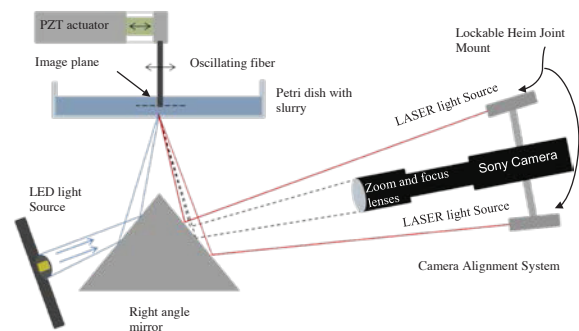


FIGURE 1. Current particle manipulation experimental apparatus.

Figure 2 shows a photograph of the experimental facility. The three legged bridge-type frame is constructed from aluminum and used as a rigid mount for the oscillator drive mechanisms, described in the next section. The LED and right angle mirror are both attached to an adjustable leg which enables alignment of the image plane on the mirror as well as directing the light from the LED to the flow region. The camera positioning stage, the fluid adjustment platform, and the LED and mirror height adjustment move independently relative to each other to provide maximum alignment flexibility.

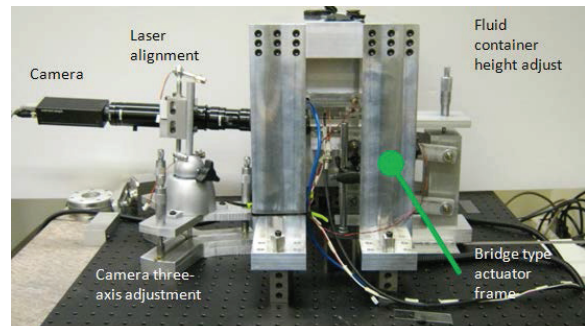


FIGURE 2. Photograph of Particle manipulation experimental apparatus

Oscillation mechanism

A flexure-based vibration mechanism has been designed and developed to induce motion to the particles inside the slurry. Figure 3 shows an image of the drive flexure. The flexure and mounting platform are made out of aluminum. The AC voltage applied to the piezo actuator will oscillate a cantilevered rod along the actuator motion axis shown in the figure with maximum amplitudes corresponding to cantilever resonances given by

$$f_n = (\alpha_n l)^2 \frac{1}{2\pi} \sqrt{\frac{EI}{mL^3}} \quad (1)$$

Where I is the second moment of area about the bending axis, m is the mass of the cantilever, L its length, and for the first natural mode $\alpha_n l = 1.87$. To obtain efficient energy transfer to the fibers attached to the piezoelectric actuator it is necessary to drive them at the resonant frequency. A molybdenum wire of 0.5 mm diameter ($E = 329$ GPa) has been used as the oscillating fiber with first mode frequency around 270 Hz.

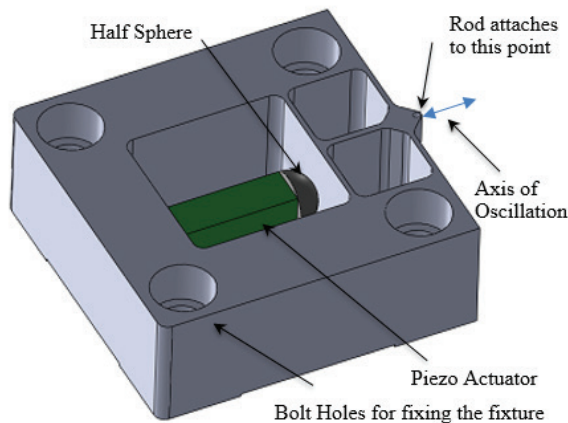


FIGURE 3. Flexure model showing the axis of oscillation that is perpendicular to the motion of piezo actuator.

To generate the drive signal at a defined frequency, a SAJI Waveform Generator (STQ2) is used (figure 4) [6]. This uses a Direct Digital Synthesizer (DDS) chip programmed with a microprocessor through Serial Peripheral Interface (SPI). The board comes with 25 MHz clock which provides 0.1 Hz resolution over 2 MHz range.

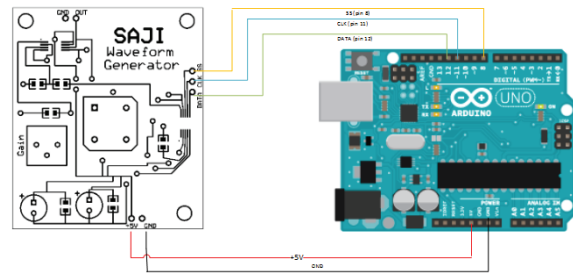


FIGURE 4. SAJI Wave Generator connection to Arduino microprocessor via SPI communication

IMAGE ANALYSIS AND PARTICLE TRACKING

Illumination

Illumination of particles plays an important role in recording high quality movies with the camera. Illumination uses combinations of 40W RGBW LED high brightness sources. It is necessary to prevent the redundant lights in the recorded images which make the image analysis and processing extensively complicated. More detail about the development of the optical imaging system and the techniques use to optimize it to reduce the light stray is provided in reference [7]. The process control provides trigger signals to synchronize the capture of camera frames with pulse timing of the LED's.

Digital image filtering

Image frames are filtered to remove the redundant and stationary objects that often show up as unusually bright spots (mostly settled particles on the bottom of the slide). Being stationary, they are exposed with an intensity proportional to the number of pulses per frame. As such these can be readily identified as having a value above the threshold for moving particles. Once identified, these are replaced by the intensity of neighborhood pixels. Figure 5, illustrates the filtering sequence and the result for one sample picture with the processing algorithm

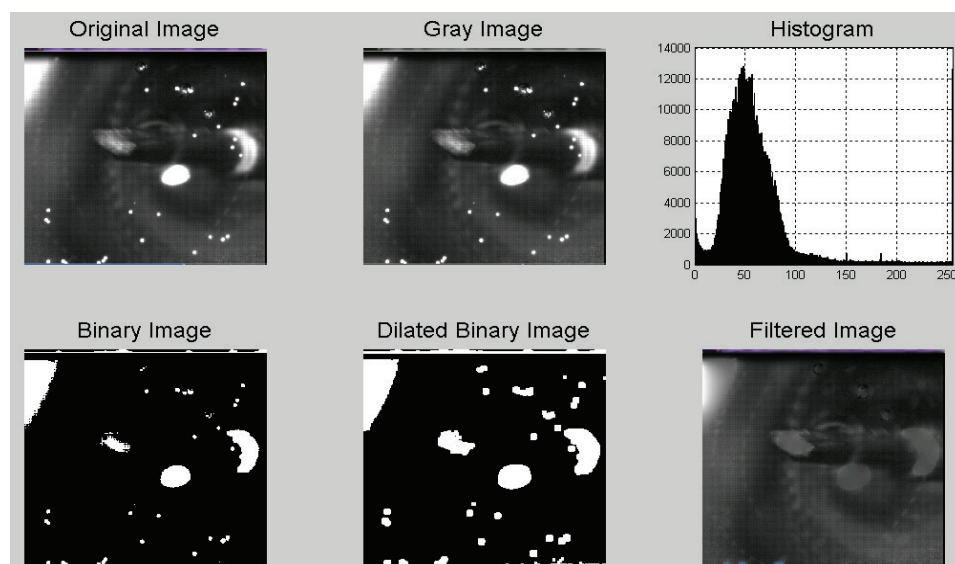


FIGURE 5. Filtering sequence showing five steps from original frame to final filtered image

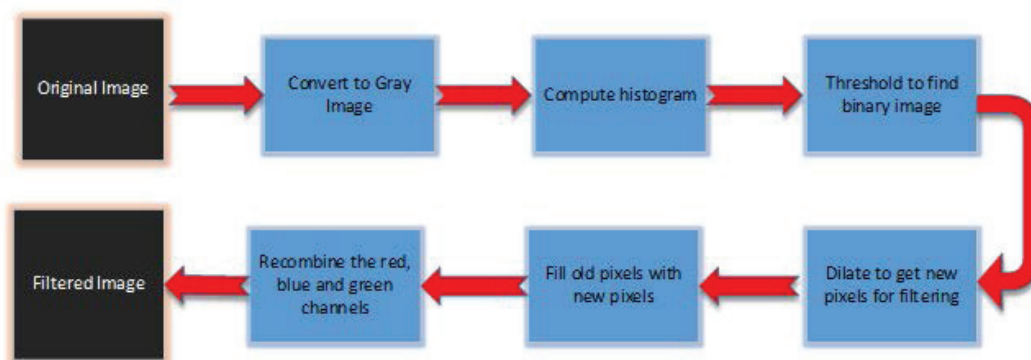


FIGURE 6. Filtering algorithm

being shown in figure 6. Figure 7 shows an example of an original image with the obscuring,

high intensity stationary particles and unwanted reflections. The image on the right is obtained after implementing the filtering algorithm.

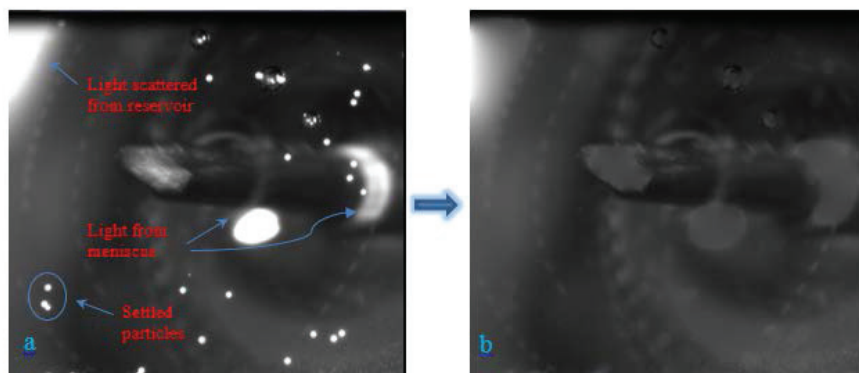


FIGURE 7. a) Original image, b) Filtered image.

RESULTS

An image of recorded videos with different LED pulsing patterns are shown in Figures 8, 9, 11, and 12. All images are obtained with the oscillating fiber in steady state condition and particles circulating about the axis of the rod. From these images, as

would be expected, the number of images of a particular particle is directly related to the number of pulses with the apparent length of the particle, for a given velocity, being related to the pulse time. Figure 8 corresponds to the LED being pulsed twice within a camera exposure. As expected, this results in apparent particle pairs, some of which are indicated with ellipses in the image.

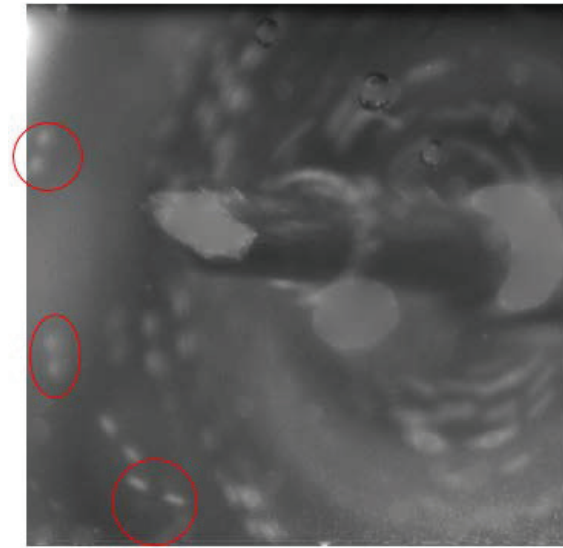


FIGURE 8. Flow fields image with 270 Hz rod frequency, 10.6 mm reservoir inner diameter, 30 ms LED on, 30 ms LED off, 30 ms camera off, 2 blinks, 120-150 μm diameter spheres.

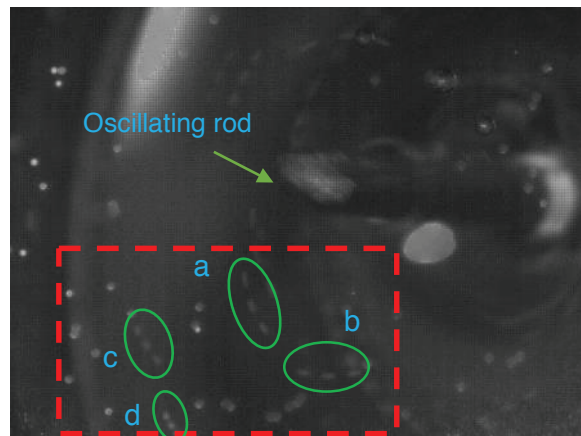


FIGURE 9. Flow fields image with 270 Hz rod frequency, 10.6 mm reservoir inner diameter, 30 ms LED on, 30 ms LED off, 30 ms camera off, 3 blinks, 120-150 μm diameter spheres.

The use of high brightness pulses with direct micro-processor control provides considerable imaging mode flexibility for the quantitative measurement of particle trajectories and velocities in the complex micro scale flow regimes in our studies.

To illustrate particle velocity extraction, a specific area in figure 9 highlighted in the red box has been analyzed. This region contains 4 particle trajectories which are labeled a,b,c and d. These particles are moving around the oscillating rod in an approximately circular path. Figure 10 shows this region after filtering the redundant scattered light and settled particles from the image. A velocity measurement program developed in Matlab is used to detect the velocity of these particles. This is obtain by finding the distance between the centroids of particles and dividing it by the LED off time, which is 30 ms in this case. As it shown in Table 1, the values of velocities are varying, trajectories closer to the rod (a and b) tend to move faster than the ones further away from it (c and d).



FIGURE 10. Trajectories of particles shown in red box in figure 9.

TABLE 1. Velocities of particle trajectories shown in figure 11.

Trajectory	a	b	c	d
Velocity (mm/s)	4.85	4.69	3.08	1.93

An example of the trajectory capability can be observed for the simple case of increasing the number of blinks within a single exposure. This is shown for 10 and 20 'blinks' in figures 11 and 12 respectively for which individual particles can be seen to trace out the paths of the streaming flow around the cylinder.

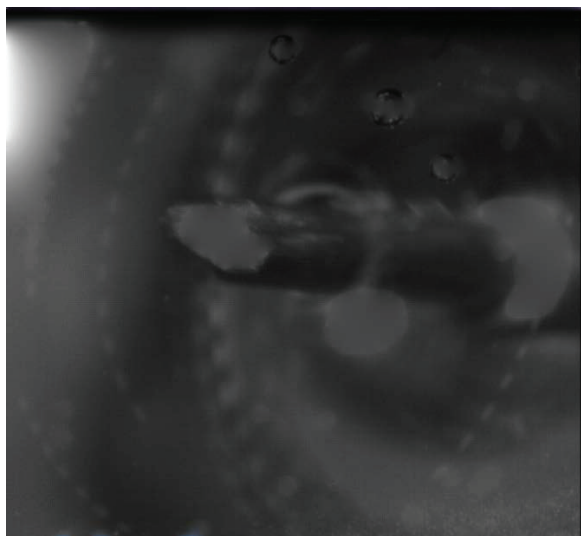


FIGURE 11. Flow fields image with 270 Hz rod frequency, 10.6 mm reservoir inner diameter, 30 ms LED on, 30 ms LED off, 30 ms camera off, 10 blinks, 120-150 μm diameter spheres.

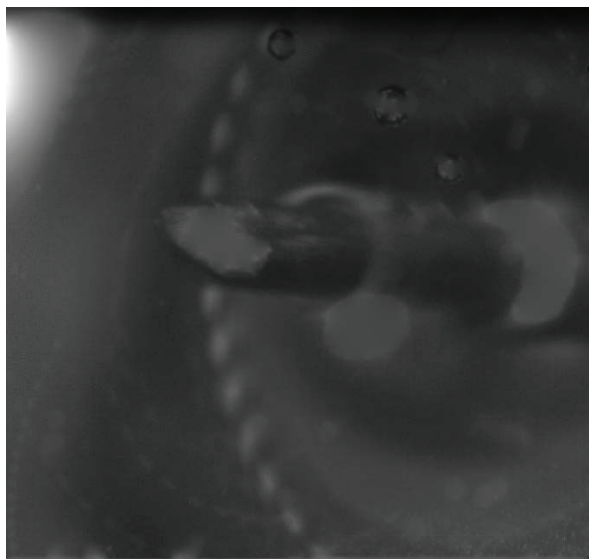


FIGURE 12. Flow fields image with 270 Hz rod frequency, 10.6 mm reservoir inner diameter, 30 ms LED on, 30 ms LED off, 30 ms camera off, 20 blinks, 120-150 μm diameter spheres.

CONCLUSION

One of the main attributes of this technique is the flexibility of illumination. Multiple combinations of the LED on and off time can be controlled to optimize measurements in regions with widely different velocities. This facility enables multi-pulsed PIV for encoding the particles' trajectories that, to our knowledge, has not been

implemented in PIV researches [8]. A major motivation for this effort is to enable the separation and tracking of multiple particles in these complex micro scale flows.

FUTURE WORK

Use of the develop facility for the optimization of multi-pulsed PIV is in the early stages and currently being evaluated to determine the robustness of the proposed techniques. This study currently requires the development of computationally demanding Monte Carlo analysis of images as well as developing mathematical models to better quantify the ability of encoded illumination to separate individual particles in 'noisy' and complex flow environments. An outcome of this study is aimed at the determination of optimal encoding conditions as well as estimates of the likelihood of extracting true particle behavior from noisy datasets.

Research of the micro-particle flow patterns and comparison of these to theoretical models [9] have been initiated in recent years. A more systematic study to assess the influence of boundary conditions, particle geometry and density with multiple micro-fiber oscillators are planned.

ACKNOWLEDGEMENT

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REFERENCES

- [1] J. S. de Bono, H. I. Scher, R. B. Montgomery, C. Parker, M. C. Miller, H. Tissing, G. V. Doyle, L. W. W. M. Terstappen, K. J. Pienta, and D. Raghavan. Circulating tumor cells predict survival benefit from treatment in metastatic castration-resistant prostate cancer. *Clinical Cancer Research*, 14:6302–6309, 2008.
- [2] Howard S.C., Chesna J. W., Mullany B. A. and Smith S.T., Subcomponent developments for a vortex machining test facility, *Proc. ASPE.*, 53, 2011
- [3] Nowakowski, B. K., Smith, S. T., Mullany, B. A., and Woody, S. C., "Vortex Machining: Localized Surface Modification Using an Oscillating Fiber Probe," *Mach. Sci. Technol.*, 13, pp. 561–570, 2009.

- [4] Howard, S., Chesna, J. W., Mullany, B., and Smith, S. T., "Observations During Vortex Machining Process Development," Proceedings of ASME, Notre Dame, 2012.
- [5] Howard, S., Mullany, B., Smith, S., "Design and Implementation of a High-Power Machining Facility for Investigations in Vortex Machining," Proceedings of 13th International Conference of Euspen, Bedford, pp. 213–216, 2013.
- [6] www.sajico.com
- [7] Kafashi, S., Eldredge, J., Thousands, J., Chong, K., Kelly, S.D., Smith, S.T. "Development of Experimental Facilities for Investigations of Microscopic Mapping of Fluid Velocities" 29th Annual Proceedings of American Society for Precision Engineering, ASPE2014, November 9-14, 2014, Boston, Massachusetts, USA.
- [8] Adrian, R. J. "Twenty years of particle image velocimetry," Experiments in Fluids (2005) 39: 159–169
- [9] Chong, K., Kelly, S., Smith, S., Eldredge, J., "Internal particle trapping in viscose streaming," Phys. Fluids 25, 033602 (2013); doi: 10.1063/1.4795857.

MOBILITY ANALYSIS OF FLEXURE SYSTEMS USING LOAD FLOW VISUALIZATION

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INTRODUCTION

A flexure system refers to an assembly of several spatially oriented slender elastic elements that determine the mobility between two rigid bodies. Each flexure element restricts motion along a specified direction (called constraint direction), while permitting motion in other directions. Thus, a flexure system will have a degree of freedom (DoF), which is the number of independent directions in three-dimensional space along which unrestricted motion is permitted. Determining these DoFs constitute analysis of flexure mobility. Consequently, the crux of a flexure synthesis problem is to determine the number of flexure building blocks and their spatial orientation to effect a degree of freedom (DoF) between two rigid bodies. The ability to perform quick mobility analysis accelerates the synthesis process of flexure systems.

This paper presents a kinetostatic framework for analyzing mobility of wire flexure systems. The concept of load flow is used to capture the deformation behavior of a flexure mechanism. Load flow enables visualization of the path taken by the applied forces through the geometry of a flexure system. The framework enables identifying constraint and freedom directions by a juxtaposition of load flow with the orientation of the mechanism geometry. This concept is analytically captured in a novel matrix-based technique to automate the mobility analysis through a metric that captures the overall orientation between the material geometry and load flow. The usefulness of this method is demonstrated in the mobility analysis of examples such as a fluid-filled fiber-reinforced elastomeric cylinder ubiquitous in nature, where traditional methods may be extremely involved. While the current framework is limited to wire flexures, it can be extended to other flexure types.

KINETOSTATIC FORMULATION FOR LOAD-FLOW VISUALIZATION

Load flow is a vector field of fictitious forces that seem to flow from origin to the fixed supports, and

through the material that constitutes the mechanism [1]. The nature of load flow in a member characterizes its functionality towards the overall deformation behavior of the mechanism. The authors' previous work [2] have quantified load flow in terms of transferred forces, which are the reaction forces at each member in the geometry. This provides a ready means to quickly evaluate load flow and generate visual insight in flexure systems modeled by slender elements such as beams and frames. Transferred forces will be used in this paper for mobility analysis of flexure systems.

VISUAL INTERPRETATION OF CONSTRAINTS AND FREEDOMS

In this section, we propose a hypothesis that will enable visual identification of freedom and constraints in wire flexure systems.

Hypothesis: In a wire flexure system, along a constraint direction, there exists at least one load path between the input and fixed ports where the transferred load is predominantly a pure force oriented along the axis of its constituent wire flexures.

In constraint-based design this hypothesis is equivalent to representing the wire flexure as a constraint line (or wrench) [3] [4]. From the definition of transferred load, axial load flow indicates that the input deflection in the constraint direction leads to negligible deflection along every part of the geometry. Thus, finding all input displacement directions that lead to axial load flow determines the constraint directions. Every other load flow direction leads to a freedom. Such a hypothesis presents necessary and sufficient conditions to identify constraints and freedom directions and is consistent with similar definitions from constraint-based design [3]. Fig.1 illustrates the freedom and constraint directions of a parallelogram flexure within a planar framework.

Overconstrained Flexures

A flexure system is exactly constrained if by removing any one flexure element, its constraint

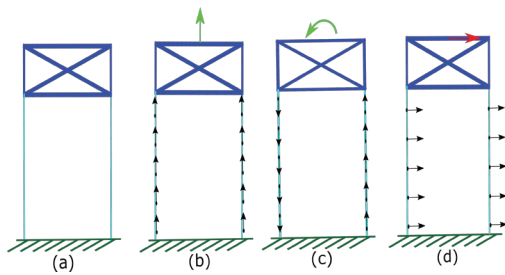


FIGURE 1. (a) A parallelogram flexure consisting of two wire flexures in parallel. Load flow for (b) axial displacement (c) in-plane twist and (d) transverse displacement.

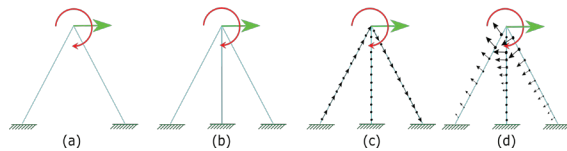


FIGURE 2. Load flow in over constrained parallel flexures. (a) A triangle flexure with a freedom (red) and constraint (green) marked, (b) an over constrained triangle flexure, (c) its load flow in the constraint direction and (d) its load flow in the freedom direction.

behavior changes. However, in some examples, flexure systems can be overconstrained. Consider the example of a triangle flexure system with two intersecting flexures as seen in Fig.2a. Adding an additional constraint element as in Fig.2b is not supposed to change the constraint nature but over-constrains the system. This effect is captured by the load flow as shown in Fig.2c, where a unit X translation is applied as input. The two flexure elements bear the load flow just as in the exactly constrained case. The third flexure carries very little load flow within it, albeit in the transverse direction. Thus, the predominant load flow direction in two load paths is along its axis, signifying a constraint direction as stated in the hypothesis.

Serial Flexures

In a series flexure there exists at least two wire flexures in a serial combination. Consider one such series flexure shown in Fig.3. According to constraint-based design this system has three degrees of freedom in a planar case. However, when evaluating the load flow in the direction shown in Fig.3a, it is seen that one of the wire flexure elements has axial load flow (implying a

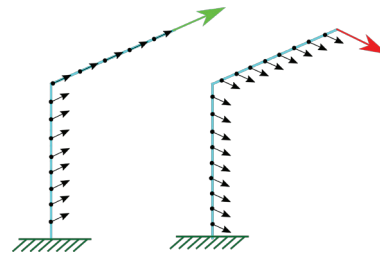


FIGURE 3. Load flow for a series flexure in the constraint and freedom directions.

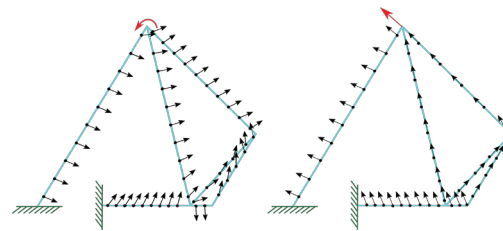


FIGURE 4. Load flows indicating freedom directions for a 2D hybrid flexure

constraint) while the other element has transverse load flow (implying a freedom). Nevertheless, there does not exist a single load path between the input and ground where the transferred load is predominantly axial. Thus the claims made in the hypothesis will not classify this case as a constraint. Additionally, input displacements applied in every other direction yields transverse load flow in both flexure members.

Hybrid Flexures

Hybrid flexure systems have a complex topology seen as a combination of serial and parallel systems. Consider the hybrid system shown in Fig.4. Upon analyzing the topology, it was seen that a pure Z rotation about the input yields transverse load flow in most of the topology indicating a freedom direction. Furthermore, there exists one translational direction as shown in Fig.4 where the transferred load is transverse indicating another degree of freedom. For an input translation in any other direction, the load flow is axial indicating a constraint. This is a two DoF flexure system. Such a visualization is insightful for determining the mobility behavior for complex hybrid topologies.

Spatial Flexures

For a spatial flexure the load flow at a point would have three corresponding force components and

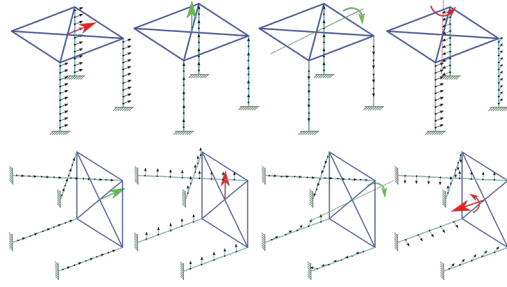


FIGURE 5. Load flows indicating freedom directions for a 2D hybrid flexure

three moment components. In three dimensions, displacements are denoted using screw theory representation as twist vectors and forces as wrench vectors. For a wrench, a scalar quantity called pitch denotes the ratio between moment and force magnitudes applied along the wrench axis. Thus, in three dimensional case, our hypothesis requires a load path whose transferred load is a zero pitch wrench acting on the flexure member and oriented along its axis. Fig.5 shows a few of the freedom and constraint directions of two spatial flexure examples and their corresponding load flows.

FRAMEWORK FOR AUTOMATED IDENTIFICATION OF CONSTRAINTS AND FREEDOMS

We propose a framework that assertively identifies constraint and freedom directions for a flexure system without the need for user intervention. The framework considers the flexure system as an ensemble, and uses collective load flow information to identify freedom and constraint directions. This automates the mobility analysis process as the user will not require to perform manual algebraic reduction. Consequently, a Load Flow Component metric (LFC) is proposed to capture the component of the load flow along the wire flexure geometry and is defined as

$$m_{LFC} = \frac{\int_0^l (\mathbf{f}(s) \cdot \hat{\mathbf{n}}(s))^2 ds}{\int_0^l ds} \quad (1)$$

Where m_{LFC} is the metric value, l is the total length of all wire flexures in the geometry s is the length parameter, $\hat{\mathbf{n}}(s)$ is the unit vector along the flexure axis at position s , $\mathbf{f}$ is the transferred force at position s and $\cdot$ denotes the vector dot product.

Matrix-based Implementation

For ease in practical implementation, it is convenient to evaluate the metric in discrete finite elements rather than in a continuous form. The steps

involved are detailed below along with an example to illustrate the methodology.

I. Mesh the geometry

The flexure geometry is divided into a number of finite elements. In each element, the transferred load is evaluated. However, load flow values at certain regions such as the junction where two flexures intersect and the point of input may not be useful as they are marred by local effects. Values further away from these regions are more indicative of the actual behavior and may be considered in the analysis.

II. Evaluate load flow in six independent basis directions

Spatial motion can be represented by six parameters and thus require six independent bases. It is with respect to these bases that the final freedom and constraint directions will be determined. In the presented examples, plucker bases, which represent any spatial screw motion are used and to determine the load flow in each element of a flexure limb, a unit displacement is applied at the input along the six bases vectors. The load flow for each case is then used to assemble the Load Flow Component matrix.

III. Assemble the Load Flow Component (LFC) matrix

For each element, the component of the transferred force along the direction of the flexure axis (dot product between force and geometry) is evaluated and stacked column-wise. This is repeated for all the bases yielding a $n \times 6$ matrix, where n denotes the number of elements in all the flexure limbs. Elements that account for aberration in load flow close to flexure junctions are not included in the matrix assembly. For normalization, load flow values are multiplied by the square root of the ratio between the elemental length and the total wire flexure length. The load flow values are also normalized by the magnitude of the maximum recorded transverse force from the six bases.

$$\mathbf{L} = \begin{bmatrix} \mathbf{f}_1^1 \cdot \hat{\mathbf{n}}_1 \sqrt{\frac{l_1}{l}} & \mathbf{f}_1^2 \cdot \hat{\mathbf{n}}_1 \sqrt{\frac{l_1}{l}} & \mathbf{f}_1^3 \cdot \hat{\mathbf{n}}_1 \sqrt{\frac{l_1}{l}} & \mathbf{f}_1^4 \cdot \hat{\mathbf{n}}_1 \sqrt{\frac{l_1}{l}} & \mathbf{f}_1^5 \cdot \hat{\mathbf{n}}_1 \sqrt{\frac{l_1}{l}} & \mathbf{f}_1^6 \cdot \hat{\mathbf{n}}_1 \sqrt{\frac{l_1}{l}} \\ \mathbf{f}_2^1 \cdot \hat{\mathbf{n}}_2 \sqrt{\frac{l_2}{l}} & \mathbf{f}_2^2 \cdot \hat{\mathbf{n}}_2 \sqrt{\frac{l_2}{l}} & \mathbf{f}_2^3 \cdot \hat{\mathbf{n}}_2 \sqrt{\frac{l_2}{l}} & \mathbf{f}_2^4 \cdot \hat{\mathbf{n}}_2 \sqrt{\frac{l_2}{l}} & \mathbf{f}_2^5 \cdot \hat{\mathbf{n}}_2 \sqrt{\frac{l_2}{l}} & \mathbf{f}_2^6 \cdot \hat{\mathbf{n}}_2 \sqrt{\frac{l_2}{l}} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \mathbf{f}_i^1 \cdot \hat{\mathbf{n}}_i \sqrt{\frac{l_i}{l}} & \mathbf{f}_i^2 \cdot \hat{\mathbf{n}}_i \sqrt{\frac{l_i}{l}} & \mathbf{f}_i^3 \cdot \hat{\mathbf{n}}_i \sqrt{\frac{l_i}{l}} & \mathbf{f}_i^4 \cdot \hat{\mathbf{n}}_i \sqrt{\frac{l_i}{l}} & \mathbf{f}_i^5 \cdot \hat{\mathbf{n}}_i \sqrt{\frac{l_i}{l}} & \mathbf{f}_i^6 \cdot \hat{\mathbf{n}}_i \sqrt{\frac{l_i}{l}} \end{bmatrix} \quad (2)$$

IV. Obtain the Eigen values of the L matrix to determine the constraint and freedom directions:

Let $\mathbf{X}$ be a 1×6 vector indicating a particular spatial direction. If this direction is a freedom, then it has to yield a zero value for the LFC metric. The constrained optimization routine to minimize the L matrix may be formulated as shown below

$$\min(\mathbf{X}^T \mathbf{L}^T \mathbf{L} \mathbf{X}), \quad \mathbf{X}^T \mathbf{X} = 1 \quad (3)$$

The solution to this minimization problem reduces to a Eigenvalue formulation of the form

$$\mathbf{L}_{sym} \mathbf{v} = \lambda \mathbf{v} \quad (4)$$

For a freedom Eq.(4) reduces to

$$\mathbf{L}_{sym} \mathbf{v} = 0 \quad (5)$$

In the above equation $\mathbf{v}$ is the eigen vector, λ 's are the eigen values of $\mathbf{L}_{sym} = \mathbf{L}^T \mathbf{L}$, which is a symmetric positive semidefinite matrix. Zero eigen values occur only when transferred forces are perpendicular to the flexure axis throughout the geometry. However, practically eigen values for a freedom direction will have very small values that are several orders of magnitude lower than the eigen values for a constraint direction. The corresponding eigen vectors identify the freedom directions and if there are more than one freedom directions, the eigen vectors would span a freedom space.

RESULTS

To demonstrate the efficacy of the presented framework, certain flexures were chosen for their complexity, and inability to be intuitively analyzed. However, they have been characterized previously with the aid of existing tools such as constraint-based design, FACT and screw-theory algebra. The examples include parallel, hybrid and serial flexures systems of increasing complexity in terms of number of flexure elements and their orientation. Additionally, the section presents examples that have not been characterized by any known means in prior literature. One such example is a building block in soft robotics known as a fluid filled fiber reinforced elastomeric cylinder.

Analysis of conventional 3D flexure systems

Fig.6 shows the examples that were analyzed using the presented framework. In the first example,

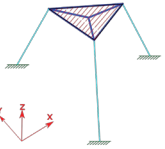
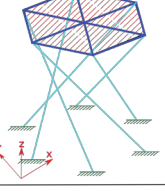
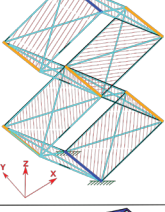
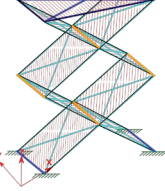
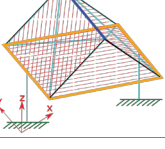
Geometry	Eigen Values	Eigen Vectors	No. of Freedoms
	$\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0.4166 \\ 0.4166 \\ 1.6694 \end{bmatrix}$	$\begin{bmatrix} 0 & -0.707 & 0.002 & 0 & 0.707 & 0 \\ 0 & -0.002 & -0.707 & 0.707 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & -1 \\ 0 & -0.002 & -0.707 & 0 & 0 & 0 \\ 0 & 0.707 & -0.002 & 0 & 0.707 & 0 \\ 1 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}$	3
	$\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0.9924 \\ 0.9929 \\ 2.1856 \end{bmatrix}$	$\begin{bmatrix} 0.346 & -0.289 & 0.216 & 0.627 & 0.597 & -0.001 \\ 0.130 & 0.379 & 0.299 & -0.597 & 0.627 & -0.002 \\ 0.135 & 0.061 & -0.136 & 0.001 & -0.002 & -0.980 \\ -0.494 & -0.318 & -0.636 & -0.117 & 0.486 & -0.001 \\ 0.407 & -0.762 & 0.064 & -0.486 & -0.118 & 0 \\ 0.660 & 0.297 & -0.661 & 0 & 0 & 0.201 \end{bmatrix}$	3
	$\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0.0434 \end{bmatrix}$	$\begin{bmatrix} -0.521 & -0.01 & -0.151 & 0.840 & 0.018 & 0 \\ 0 & 0.541 & 0.022 & 0.029 & -0.840 & 0 \\ 0.029 & 0.008 & 0.980 & 0.194 & 0.037 & 0 \\ 0 & 0.004 & -0.841 & 0.026 & 0.007 & -0.541 \\ 0 & -0.853 & 0.004 & 0.125 & -0.506 & -0.012 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}$	5
	$\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0.1147 \\ 0.2331 \end{bmatrix}$	$\begin{bmatrix} -0.600 & 0.005 & 0.019 & 0.800 & 0 & 0 \\ 0.003 & -0.707 & -0.003 & 0.007 & 0 & -0.707 \\ 0.003 & 0.005 & -1 & 0.026 & 0 & 0 \\ 0 & -0.003 & 0.707 & 0.003 & 0.007 & -0.707 \\ 0.1147 & -0.800 & -0.009 & -0.018 & -0.599 & 0 \\ 0.2331 & 0 & 0 & 0 & 0 & -1 \end{bmatrix}$	4
Geometry	Eigen Values	Eigen Vectors	No. of Freedoms
	$\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0.4611 \end{bmatrix}$	$\begin{bmatrix} 0 & 0.176 & 0 & 0.984 & 0 & 0 \\ 0.654 & 0 & 0 & 0 & -0.756 & 0 \\ 0 & 0 & 0 & 0 & 0 & -1 \\ 0 & -0.756 & 0 & 0 & 0 & -0.654 \\ 0 & 0 & -0.984 & 0 & 0.176 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \end{bmatrix}$	5

FIGURE 6. Dof determination of 3D flexure systems. Blue- Rigid members, Cyan- Flexure limbs, Orange- Intermediate bodies

the stage is supported by three flexure members that converge at a point above the stage. Such a geometry possesses three DoFs that are comparable to three pure rotations about that point. These rotations are read as the columns of the eigen vector matrix that correspond to the smallest eigen values (first three columns). The second example has the stage supported by six flexure members that approximately takes the geometry of a circular hyperboloid. This system has three screw freedom directions.

The third and fourth examples represent hybrid flexures [5]. The synthesis and analysis of hybrid systems have been extensively developed by Hopkins with numerous supporting examples of

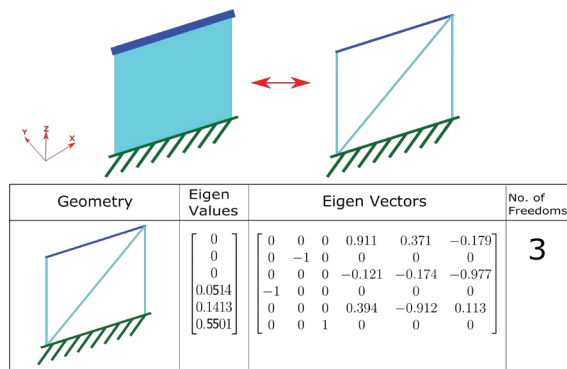


FIGURE 7. Analysis of a wire flexure system equivalent in mobility to a sheet flexure

which examples three and four are a part. While Hopkins modeled the systems using sheet flexures, we have used a system wire flexures to accurately capture the freedoms and constraints of a sheet flexure. This is seen in the analysis conducted in Fig.7 and also indicated by Hopkins [6].

It must be mentioned that to a novice designer, the analysis of such a flexure would be highly non-intuitive. It involves the breakdown of hybrid flexure systems into conventional components, which are then analyzed using existing FACT shapes. Depending upon the user's experience this can become lengthy and involved. Our framework accepts the geometry unaltered and returns relevant results. For example three, we obtain five Dofs (five screws) with rotation along z -axis being constrained. Similarly, example four yields four Dofs (four screws).

Example five represents a serial flexure system. It consists of two parallel flexure systems that are concatenated. The first system has wire flexures and the second system has sheet flexures. Our framework yields five Dofs with the system being constrained for vertical translation. It is important to note that all these results are in perfect agreement with those obtained by Hopkins.

Fiber-Reinforced Elastomeric Enclosure

Fiber-Reinforced Elastomeric Enclosures (FREEs) are popular soft actuators for robots. It consists of two families of equal and opposite helical fibers wound on the hollow elastomeric cylinder as shown in Fig. 8. The top cap of the cylinder is rigid, while the bottom cap is constrained. The fibers can be modeled as a network of spatial wire flexures. While a fiber

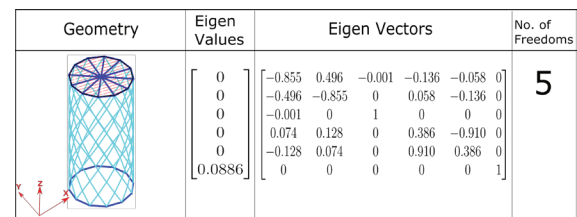


FIGURE 8. Dof determination of FREE

is inherently stiff in tension, stiffness in all the other directions is provided by the elastomer matrix and fluid pressure. The fibers are aligned at 54.74° with respect to the flexure axis. Such a configuration, upon pressurization with fluids does not actuate but stiffens. Due to hydrostatic pressure, the cross-section of the cylinder is constrained against distortion or kinking. To account for this each flexure intersection point is constrained against a rotation about the cylinder axis or Z -rotation. It is now required to analyze directions where the relative stiffness of the cylinder is maximum.

The framework was applied to this problem by dividing each flexure into five elements, evaluating load flow and assembling the $\mathbf{L}$ and $\mathbf{L}_{sym}$ matrices. The eigen values of these matrices are also shown in Fig.8. It is seen that the geometry has five near zero eigen values, and corresponding freedom spaces. The only direction along which there is a non-zero eigen value is a pure rotation along the axis of the cylinder. Thus, such a FREE is considerably stiff against torsion and does not permit this motion unless the cross-section distorts against the pressure. This result concurs with the general observation in Vogel (2003) [7]. Furthermore, it concurs with the widely accepted practice in industry of reinforcing hoses with such a fiber arrangement to stiffen the cross-section against twisting while simultaneously allowing deflection in every other direction.

CONCLUSION

Mobility analysis is an important step in synthesizing spatial flexure systems. It involves determining freedom and constraint directions, which are qualitative descriptions of the deformation behavior. While recent literature stresses on projective geometry based visualization or analytical method involving screw-theory, the work presented in the paper combines both visual insight and analytical rigor. The basis of the analysis framework presented involves a kinetostatic load

flow visualization and characterization. Two main techniques are proposed:

1. A visual technique for mobility analysis, where freedoms and constraints are classified based on the direction of load flow in flexures. This requires the user to manually post-process load flow directions.
2. A matrix-based technique for automated identification of flexure mobility. This characterizes load flow in the flexure geometry as an ensemble within a matrix and applies standard reduction techniques to determine freedom and constraint directions.

The advantage of the proposed formulation is its ability to automate flexure mobility analysis with little or no user intervention. Thus, irrespective of the complexity of the flexure system its freedom and constraint directions can be evaluated from at the most six matrix-based finite element analysis, requiring just a few seconds (assuming standard computing capabilities). The automated analysis framework can serve as an excellent platform to implement FACT-based synthesis of spatial flexures, thereby reducing its learning curve. Towards this, future work will map the mobility results from the proposed method to FACT shapes. Furthermore, in future work guidelines will be established to compare qualitatively the load flow behavior of two flexure geometries that have similar mobility. Extension to large deformations can also aid in identifying geometries with robust constraint properties. Extension to include other flexure types will also be investigated.

ACKNOWLEDGMENTS

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REFERENCES

- [1] Harasaki H, Arora JS. New concepts of transferred and potential transferred forces in structures. *Computer methods in applied mechanics and engineering*. 2001;191(3):385–406.
- [2] Krishnan G, Kim C, Kota S. A Kinetostatic Formulation for Load-Flow Visualization in Compliant Mechanisms. *Journal of Mechanisms and Robotics*. 2013;5(2):021007.
- [3] Synthesis of multi-degree of freedom, parallel flexure system concepts via Freedom and Constraint Topology (FACT) Part I: Principles. *Precision Engineering*. 2010;34(2):259 – 270.
- [4] Synthesis of multi-degree of freedom, parallel flexure system concepts via freedom and constraint topology (FACT). Part II: Practice. *Precision Engineering*. 2010;34(2):271 – 278.
- [5] Hopkins J. Designing hybrid flexure systems and elements using freedom and constraint topologies. *Mech Sci*. 2013;4(2):319–331.
- [6] Hopkins JB. A Visualization Approach for Analyzing and Synthesizing Serial Flexure Elements. *Journal of Mechanisms and Robotics*. 2015;7(3):031011.
- [7] Vogel S. *Comparative Biomechanics*. Princeton, New Jersey: Princeton University Press; 2003.

DESIGN OF SPINDLE SUPPORTED BY WATER HYDROSTATIC BEARINGS

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INTRODUCTION

Recently, the demand for improving machining accuracy of precision parts including optical components and the semiconductor parts is increasing. Furthermore, the developments of the ultra-precision machine tools that do not use oil as lubricating fluid in clean environments are also required.

In many cases, low viscosity oil or air is used for lubricating fluid for the hydrostatic bearings [1, 2] of the ultra-precision machine tools. On the other hand, water is an alternative fluid for the lubricating fluid from several reasons. For instance, the water lubrication is advantageous for the spindle applications of the ultra-precision machine tools, because water is the incompressible fluid with low viscosity. Furthermore, water is the best choice for achieving better thermal stability of the spindle.

However, it should be noted that the precise design of the water hydrostatic bearings is particularly needed compared with the oil hydrostatic bearings. Because of the low viscosity of water, the bearing gap has to be designed to be small. For the bearing designs of the ultra-precision machines, a careful attention should be paid to determine the critical dimensions of the spindle in order to achieve higher bearing stiffness. In addition, suitable material selections are significantly important for the water hydrostatic bearings due to a lack of lubricant properties of the water and in order to avoid rust of the parts of the spindle.

In our previous studies, a design procedure of water hydrostatic bearing to achieve highest bearing stiffness for given operating conditions has been proposed [3]. In addition, it has been confirmed that the proposed design procedure is able to design the desired bearing performances. Then, opposed pad water hydrostatic axial

bearings were developed based on the proposed design procedure [4].

Based on the previous works, a new spindle supported by water hydrostatic axial as well as journal bearings is designed and developed. Purpose of this study is to develop the spindle that is applicable to ultra-precision machine tools. In the development of the spindle, suitable material selections are also carefully considered. The developed water hydrostatic spindle is designed so that bearing stiffness of the axial and journal bearings becomes 1,000 N/ μ m and 500 N/ μ m, respectively. The rated rotational speed of the spindle is designed to be 5,000 min⁻¹. Critical mating surfaces of the rotor and inner surfaces of the casing are coated by a DLC plating.

DESIGN OF WATER HYDROSTATIC BEARING

Basic Structure and Specification of Water Hydrostatic Spindle

A basic structure and dimensions of the designed spindle are depicted in Fig. 1. Target bearing stiffness of the water hydrostatic bearings of the spindle is set to be axial bearing stiffness of 1,000 N/ μ m and journal bearing stiffness of 500 N/ μ m, as described. This specification of the bearing stiffness achieves extreme small resultant displacements due to the change in the external loads acting on the bearings during the ultra-precision machining. Specifically, the resultant displacement of the bearing can be constrained less than several nanometers due to the change in the cutting forces of 1 N or less during diamond turning. Rated rotational speed of the spindle is defined to be 5,000 min⁻¹.

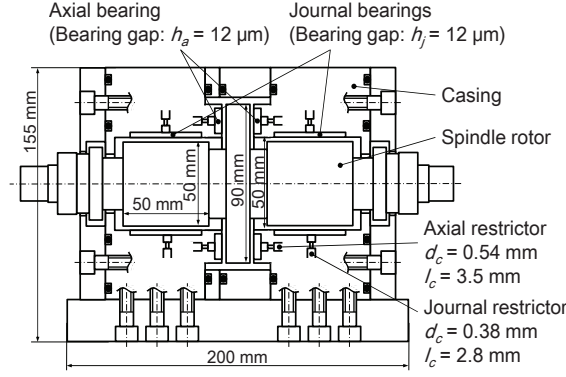


FIGURE 1. Schematic of designed spindle

NOMENCLATURE

- d_c : diameter of restrictor
 h_0, h_a, h_j, h_e : bearing gap
 l_c : length of restrictor
 m_a, m_j : number of bearing surface (= 2, 2)
 m : number of recesses (= 1)
 n : number of pads (= 1)
 p_{01}, p_{02} : non-dimensional pressure
 p_s : supply pressure
 p_r : recess pressure
 r_{a1}, r_{a2} : radius of axial bearing (= 50, 90 mm)
 r_j : radius of journal bearing (= 50 mm)
 y : displacement of bearing
 A : area of bearing pad
 $\bar{A}$: coefficient of effective area of bearing pad
 A_c : area of restrictor
 $\bar{B}$: flow coefficient of bearing pad
 K, K_0 : bearing stiffness
 L : length of journal bearing (= 50 mm)
 P_v : power loss
 T_v, T_a, T_j : viscous torque
 λ_1, λ_2 : friction factor
 μ : viscosity of water
 ρ : density of water
 τ_a, τ_j : shearing stress
 ζ_1, ζ_2 : loss coefficients at entrance and exit

Design of Opposed Pad Water Hydrostatic Axial Bearing

The water hydrostatic bearings are designed based on the proposed design procedure for the opposed pad hydrostatic bearings [5]. Based on the design procedure, the bearing gap and rated supply pressure of the axial bearing to achieve the required bearing stiffness are determined as follow.

First, the axial bearing stiffness K of opposed pad water hydrostatic bearings is represented by Eq. (1).

$$K = \frac{-3m\bar{A}Ap_s}{\alpha h_0} \left\{ \frac{\xi_1 - 2\alpha\kappa_1(1-\varepsilon)^6 - 1}{\xi_1\kappa_1(1-\varepsilon)^7} + \frac{\xi_2 - 2\alpha\kappa_2(1+\varepsilon)^6 - 1}{\xi_2\kappa_2(1+\varepsilon)^7} \right\} \quad (1)$$

Here,

$$\xi_1 = \sqrt{1 + 4\alpha\kappa_1(1-\varepsilon)^6} \quad (2-1)$$

$$\xi_2 = \sqrt{1 + 4\alpha\kappa_2(1+\varepsilon)^6} \quad (2-2)$$

In Eqns. (1), (2-1) and (2-2), α is a non-dimensional parameter given by Eq. (3).

$$\alpha = \frac{\rho p_s}{2A_c^2} \left(\frac{\bar{B}h_0^3}{n\mu} \right)^2 \quad (3)$$

The displacement of the spindle rotor in the axial direction y is represented by a non-dimensional parameter ε as Eq. (4).

$$\varepsilon = \frac{y}{h_0} \quad (4)$$

Furthermore, coefficients of the flow characteristic of the bearing restrictors κ_1 and κ_2 are given by Eqns. (5-1) and (5-2).

$$\kappa_1 = \lambda_1 \cdot c + \zeta_1 + \zeta_2 \quad (5-1)$$

$$\kappa_2 = \lambda_2 \cdot c + \zeta_1 + \zeta_2 \quad (5-2)$$

In Eqns. (5-1) and (5-2), an aspect ratio c of the short-tube restrictor is defined by Eq. (6).

$$c = \frac{l_c}{d_c} \quad (6)$$

The aspect ratio c is determined based on the proposed restrictor design method [3] to achieve the maximum bearing stiffness for given conditions. If the spindle displacement in the axial direction is assumed to be zero (i.e. $\varepsilon = 0$), κ_1 and κ_2 in Eqns. (1), (2-1) and (2-2) are the same and then represented by κ .

The optimum condition of the bearing stiffness is then defined as Eq. (7).

$$K_0 = \frac{6m\bar{A}Ap_s}{\alpha\kappa h_0} \left(\frac{1 + 2\alpha\kappa}{\sqrt{1 + 4\alpha\kappa}} - 1 \right) \quad (7)$$

Thus, critical parameters such as the supply pressure and the bearing gaps to achieve required bearing stiffness are determined using Eq. (7).

Based on Eq. (7), the axial bearing stiffness with various bearing gaps from 8 μm to 18 μm was calculated. The calculation results are shown in Fig. 2. The bearing stiffness increases by choosing smaller bearing gap and large supply pressure.

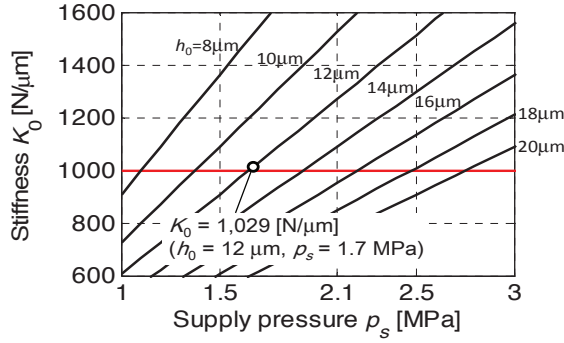


FIGURE 2. Axial bearing stiffness with optimum conditions ($\varepsilon = 0$ [-]),

As well known, if the bearing gap is small, the heat generation due to the viscosity of the lubricating fluid is remarkable. From a viewpoint of the thermal stability, thus, an optimum gap size has to be determined. Furthermore, the supply pressure is desirable to be low from a viewpoint of the reduction of the energy consumption. This is also effective to reduce heat generation of the spindle, as well.

From Fig. 2, desired bearing gap and the supply pressure can be determined as 12 μm and 1.7 MPa, respectively. As a result, the desired axial bearing stiffness of the water hydrostatic bearings of 1,000 N/ μm can be obtained. In addition to the axial bearing design, journal bearings of the spindle was designed to be the bearing gap of 12 μm and the supply pressure of 2.1 MPa.

Robust Design of Bearing Gap on Opposed Pad Water Hydrostatic Bearing

In the actual design of the precise parts, machining errors and assembly errors of the orders of the micrometers must be taken into account in order to achieve robust design. Thus, the design of the water hydrostatic bearings of the spindle is carried out by considering unpreventable dimensional errors of the bearing gaps by specifying the tolerance of 4 μm , due to the machining as well as assembly errors.

Specifically, the design is made so that the desired bearing stiffness can be obtained even the maximum error in the bearing gap exists. The design with the consideration of the errors of bearing gap is conducted as follow.

The bearing stiffness K_e in consideration of the dimensional errors is represented by Eq. (8).

$$K_e = \frac{6m\bar{A}Ap_s}{\alpha_e \kappa_e h_e} \left(\frac{1 + 2\alpha_e \kappa_e}{\sqrt{1 + 4\alpha_e \kappa_e}} - 1 \right) \quad (8)$$

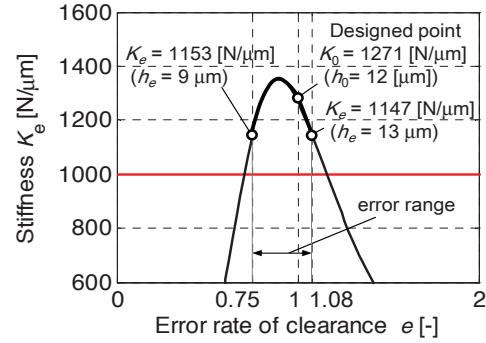


FIGURE 3. Axial bearing stiffness in consideration of dimensional error ($p_s = 2.1$ [MPa])

In Eq. (8), h_e is the bearing gap with the dimensional error. The bearing gap is defined as Eq. (9) using the error rate e .

$$h_e = h_0 \cdot e \quad (9)$$

If the error rate $e = 1$, there is no dimensional error at the bearing gaps when the spindle is assembled. In Eq. (8), α_e and κ_e are variables affected by the dimensional errors. Figure 3 shows the calculated stiffness of the axial bearing by Eq. (8). Here, the supply pressure p_s is 2.1 MPa. In Fig. 3, an ideal designed point is presented by an open circle, in this case, there is no dimensional error in the gap. Since ideal bearing gap is 12 μm , if unpreventable dimensional error in the gap is assumed to be from -3 μm to 1 μm , the error rate of e varies from 0.75 to 1.08. Based on the designed results, it is found that the bearing stiffness varies from 1,144 N/ μm to 1,153 N/ μm as presented in Fig. 3, if there is a dimensional error in the gap. Thus, the desired bearing stiffness of 1000 N/ μm is obtained regardless of the unpreventable dimensional error.

Accordingly, critical sizes of the spindle are determined so that resultant axial bearing gap becomes the size from 9 μm to 13 μm . Similarly, the bearing gap of the journal bearing was determined to be the size from 9 μm to 13 μm as well.

MOTOR SELECTION

The designed spindle is driven by an externally connected electric motor connected by a magnet coupling to the spindle. In the design, the build-in motor structure was also considered. However, the external motor structure was chosen, in order to avoid the influence of heat generation of the motor on the thermal stability of the spindle.

The spindle rotor is supported by the water hydrostatic axial and journal bearings. Accordingly, the resistance torque due to the viscous friction of water must be considered for determining the motor power for the spindle. In addition, an influence of the moment of inertia of the spindle rotor is considered as usual. Viscous torques T_a and T_j in sliding surfaces of the axial and journal bearings are calculated using Eqns. (11), (12-1) and (12-2).

$$\tau_{a,j} = \mu \frac{r_{a,j} \omega}{h_{a,j}} \quad (11)$$

$$T_a = m_a \int_{r_{a1}}^{r_{a2}} r \cdot \tau_a \cdot 2\pi r \cdot dr$$

$$= \frac{\pi m_a \mu \omega (r_{a2}^4 - r_{a1}^4)}{2h_a} \quad (12-1)$$

$$T_j = m_j \cdot \tau_j \cdot A_j \cdot r_j$$

$$= \frac{2\pi m_j \mu r_j^3 \omega L}{h_j} \quad (12-2)$$

Therefore, the total viscous torque T_v of the spindle is shown in Eq. (13)

$$T_v = T_a + T_j \quad (13)$$

Power loss P_v by the viscous torque of water hydrostatic bearings is then calculated from viscous torque and angular velocity ω by Eq. (14).

$$P_v = T_v \cdot \omega \quad (14)$$

The relationships between the spindle speed N and the viscous torque, and the power loss are shown in Figs. 4 (a) and (b).

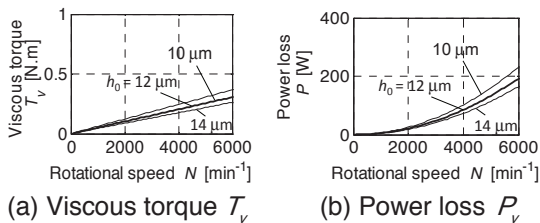


FIGURE 4. Required driving torque

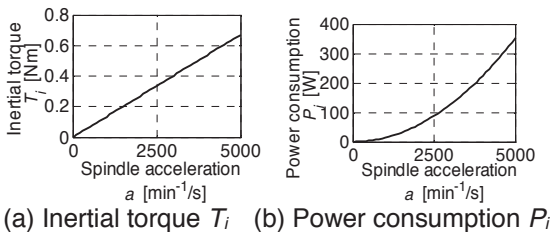


FIGURE 5. Required driving power

For instance, it can be seen that the viscous torque is 0.3 N and the power loss is 160 W at the rated rotational speed of 5,000 min⁻¹. In addition, the relationships between the acceleration a of the spindle speed and the inertial torque T_i , and the power consumption P_i are shown in Fig. 5. In Fig. 5, the spindle acceleration is presented by a rate of speed change in 1 second.

Based on the considerations, an AC servo motor which has the maximum rotational speed of 5,500 min⁻¹ and the rated torque of 1.8 Nm is chosen for the spindle. In addition, the magnet coupling that has the maximum transmission torque of 4.0 Nm is selected.

DEVELOPED SPINDLE WITH WATER HYDROSTATIC BEARINGS

The developed spindle with the water hydrostatic bearings and spindle rotor are shown in Fig. 6 and Fig. 7, respectively. DLC (Diamond-Like-Carbon) coating is plated for two main parts of the casing body and spindle rotor in order for using water-lubrication environments. Thus, it is expected to give high harder and smooth surfaces of the critical parts by the DLC coating. Table 1 summarizes the designed results of the spindle.

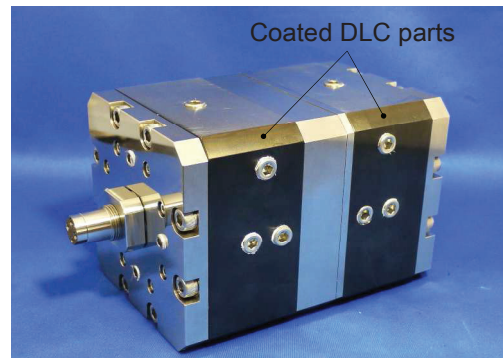


FIGURE 6. Developed water hydrostatic spindle

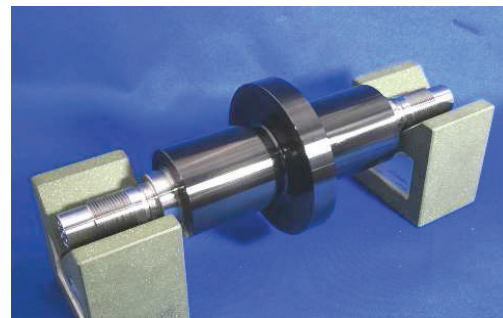


FIGURE 7. Spindle rotor with DLC plating

TABLE 1. Designed results of spindle

Type of bearing	Axial	Journal
Bearing Stiffness [N/μm]	1,000	500
Load capacity [N]	6,600	3,300
Rotational speed [min ⁻¹]	5,500	
Bearing gap [μm]	11 ± 2	11 ± 2
Supply pressure [MPa]	2.1	2.5
Flow rate [L/min]	3.7	2.0

CONCLUSIONS

In this paper, a spindle with the water hydrostatic bearings for ultra-precision machine tools was designed and developed. The water hydrostatic bearings were designed so that the axial and journal bearings become larger than 1000 N/μm and 500 N/μm. Furthermore, in the design of the water hydrostatic bearings, the influence of machining and assembly errors on the bearing stiffness was considered. Critical parts of the developed spindle were plated by DLC plating in order to improve durability in the water lubrication environment.

In the future work, the spindle performances will be investigated experimentally. Furthermore, the spindle will be also applied to the actual diamond turning tests.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] Shinshi T., Inami Y., Sato K., Shimokohbe A., An Active Aerostatic Spindle with an Axial-Positioning Actuator for Machining Ultra-Micro Scale Structure, Proc. of International Conference on Leading Edge Manufacturing in 21st Century. pp. 1015-1018, 2005.
- [2] Katto S, Kadotani M, Kitagawa T, Hirayama T, Matsuoka T, Sasaki K., Development of Pneumatic Servo Bearing Actuator for Nano-Positioning, Proc. of JSME-IIP/ASME-ISPS Joint Conference on Micromechatronics for Information and Precision Equipment, pp. 185-186, 2009.
- [3] Nakao Y., Nakatsugawa S., Komori M., Suzuki K., Design of short-Pipe Restrictor of Hydrostatic Thrust Bearings, Proc. of

ASME 2012 International Mechanical Congress and Exposition, CD-ROM 2012.

- [4] Nagasaka K., Yamada K., Yamada Y., Hayashi A, Nakao Y. Influence of Water Pressure on Stiffness of Thrust Bearing of Water Hydrostatic Spindle. Proc. of JSME 2014 Manufacturing and Machine Tool Conference, pp. 107-108, 2014 (in Japanese).
- [5] Nakao Y., Suzuki K., Yamada K., Nagasaka K., Feasibility Study on Design of Spindle Supported by High Stiffness Water Hydrostatic Thrust Bearing, International Journal of Automation Technology 8(4), pp. 530-538, 2014.

Design of Compact Multi-DOF Ultrasonic Spherical Actuator

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INSTRUCTIONS

Recently, multi degree of freedom actuator is required in many industrial applications such as robotics, precision manufacturing, vision system, and medical applications. For this reasons, the study of actuators to perform multi degree of freedom movements has increased.

Conventional multi degree actuators achieved multi degree of motions by connecting several single degree actuators in serial or parallel. These actuators have some drawbacks such as complex structure, massive body and backlash by using several links and reduction gears. These massive actuators are not applicable in the several industries such as robotics, vision systems, and medical systems.

In order to resolve these problems, the study of spherical motors to perform multi degree of freedom movements in one unit has increased.

In this study, a novel 3-DOF spherical actuator using ultrasonic motor principle is proposed. It is expected to have high torque, braking system, compact size.

PRINCIPLE OF ULTRASONIC MOTOR

The ultrasonic motor is one of interesting actuators. As shown in the Figure 1, ultrasonic motor (USM) is composed of stator, piezoelectric materials and rotor. USM utilizes the vibration of the elastic body (stator) in the ultrasonic frequency band caused by the reverse piezoelectric effect of piezoelectric materials. The actuation principle of ultrasonic motor is the friction drive between stator and rotor. The piezoelectric materials generate vibration in ultrasonic frequency and the surface of stator which is adhered to piezoelectric material makes the elliptical motion.

As ultrasonic motor is based on the friction force, the advantages of ultrasonic motor are large

torque, braking system, high responsiveness and compactness.

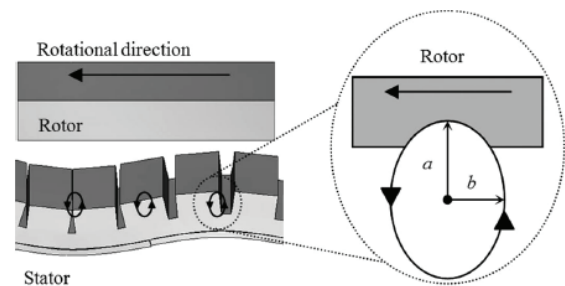


FIGURE 1. Elliptical motions on the surface of the stator

Traveling Wave

The important part of ultrasonic motor is the travelling wave generation. There are many methods to generate traveling wave in an elastic body. In this study, the superposition method of the two standing wave with phase difference is used.

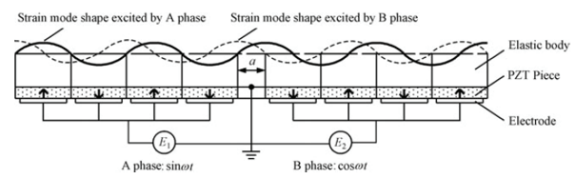


FIGURE 2. Traveling wave in an elastic body

As shown in the Figure 2, PZT pieces divided into two groups with the gap a . The voltage to A-phase electrode is E_1 and B-phase electrode is E_2 . The simple conditions to excite a travelling wave is

$$a = \frac{\lambda}{4}, \quad E_1 = E_2$$

CONCEPT DESIGN

The concept design of proposed spherical actuator is represented in Figure 3. The proposed spherical actuator is composed of X-Y tilt stator, rotor, and outer body. Z rotation ultrasonic motor and guide system is combined in the rotor. The rotor of the spherical actuator is in contact with X-Y tilt stator for two degree of freedom tilt (θ_x, θ_y) motion.

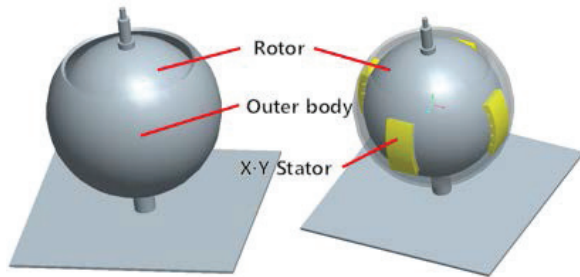


FIGURE 3. Concept of the proposed spherical actuator

X-Y axis Tilting Motion

X-Y Tilting motion of the proposed spherical actuator is actuated by curved ultrasonic motor which is shown in Figure 4. The type of proposed ultrasonic motor is linear ultrasonic motor. To contact large surface with rotor, the curved shape followed the rotor's surface line is applied to the proposed ultrasonic motor.

The curved bar-type ultrasonic motor is fixed on the outer body. In the ultrasonic motor principle, curved elastic body is vibrated by the reverse piezoelectric effect of piezoelectric materials. As shown in Figure 4, two ultrasonic motors generate X or Y tilting motion.

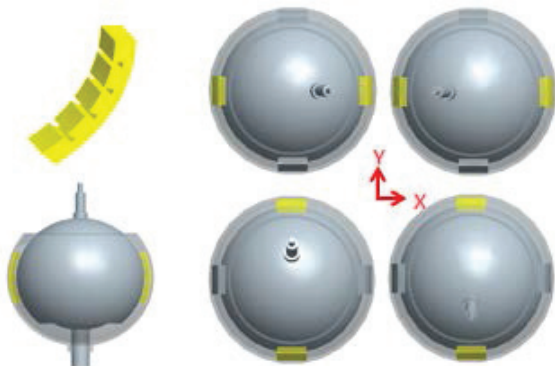


FIGURE 4. Curved bar type stator & X-Y axis tilting motion

In this system, high torque is expected in the tilting motion because two ultrasonic motors are actuated per one axis

Guide System & Z axis Rotating Motion

The proposed spherical actuator utilizes gimbal guide for guide system. A gimbal is a pivoted support that allows the rotation of an object about a single axis. The proposed gimbal guide system is shown in Figure 5. The ring shape X-axis guide rotates along with global X-axis. The gimbal body is the role of Y-axis guide. The gimbal body rotates along with local Y-axis which is relative coordinate to X-axis. In this system, each axis is decoupled to actuate tilting motion. In this decoupled tilting motion, any interference between each axis is not occurred to other axis tilting motion. For this reason, motor makes uniform performance when spherical motor generate both X and Y axis tilting motion.

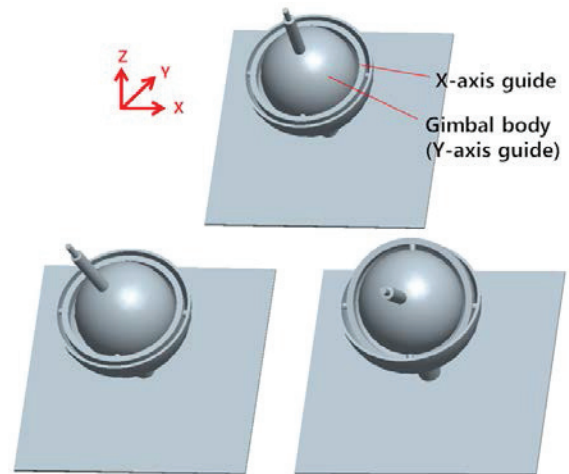


FIGURE 5. Guide system & X-Y axis decoupled tilting motion

In the proposed spherical motor, rotor and gimbal body are fixed together through one shaft. As shown in Figure 6, curved stator for Z rotating motion is fixed in gimbal body. For this reason, Z rotating motion is independent of X-Y tilting motion.

Z rotating stator has different teeth shape compared with X-Y tilting stator because of rotating direction.

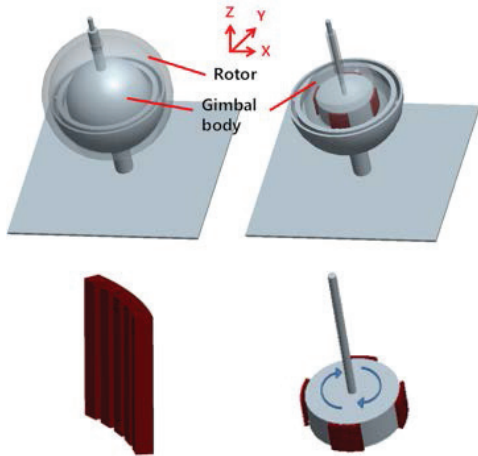


FIGURE 6. Guide System & Z axis Rotating Motion

Four Z-axis curved stators contact with one Z rotor. Similarly to X-Y-axis curved stators, Z-axis curved stator is vibrated by piezoelectric material and Z rotor is rotated by the friction between rotor and stator.

FEM SIMULATION

The stator of the curved bar-type ultrasonic motor generates travelling wave. To verify the possibility of ultrasonic motor, FEM simulation is performed. Based on the travelling wave generation method of curved plate, the proposed curved stator is shown in Figure 7.

PZT sets of opposite polarization two PZT pieces are bonded to elastic body. As shown in Figure 8, the gap between two PZT sets is a quarter of wavelength to generate travelling wave. Each PZT set is applied to sinusoidal voltage with half of wavelength phase difference.

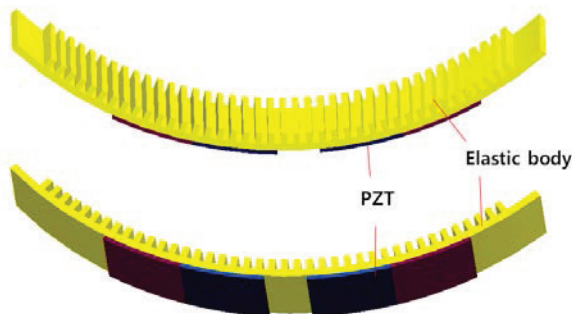


FIGURE 7. The Proposed Curved Stator

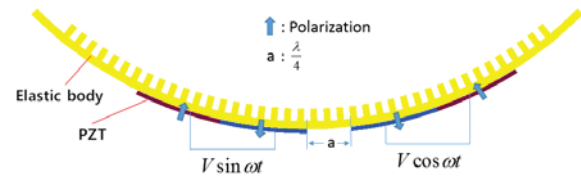


FIGURE 8. FEM Simulation Conditions

The ultrasonic motor performance like torque, rotation speed is proportional to travelling wave speed is proportional to travelling wave amplitude. In the case of ultrasonic motor, natural frequency of the curved stator is used to generate large travelling wave amplitude. For this reason, the frequency of input voltage to piezoelectric material is 22000Hz which is the natural frequency of proposed curved stator. Also, the magnitude of the voltage is 100V in this FEM simulation.

With this simulation conditions, the result of FEM simulation is shown in Figure 9. As shown in Figure 9, travelling wave of the proposed curved stator is well generated at simulation conditions. The travelling wave propagates with right to left direction and amplitude of the travelling wave is almost 0.15um.

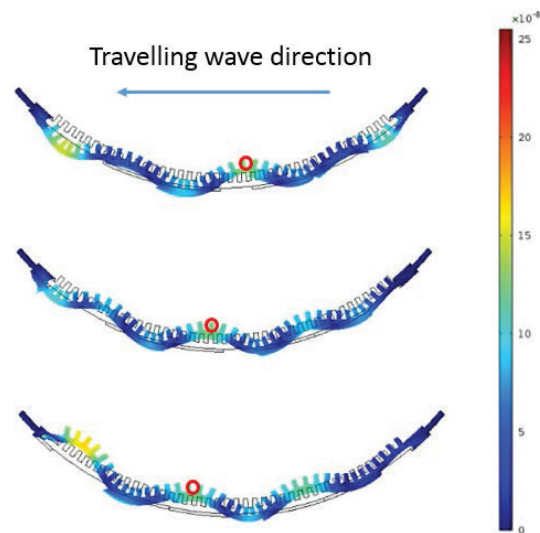


FIGURE 9. Wave Propagation of Curved Stator

As a result, it can be expected that the proposed curved stator can be a component of ultrasonic motor. Using the proposed ultrasonic motor, the multi-DOF spherical actuator can be expected to have high torque, braking system, compact size.

CONCLUSIONS AND FUTURE WORKS

In this study, a novel 3-DOF spherical actuator using ultrasonic motor principle is proposed. The proposed spherical actuator is composed of X-Y tilt stator, rotor, and outside body. Z rotation ultrasonic motor and guide system is combined in the rotor. This system is expected to have high torque, braking system, compact size.

To verify the possibility of ultrasonic motor, FEM simulation of the proposed curved stator is performed in this study. Travelling wave is well generated at 22000Hz which is the natural frequency of the curved stator and magnitude of travelling wave is almost 0.15 μ m.

In the future, stator modeling and optimization of the proposed spherical actuator will be needed for maximizing performance such as torque, angular velocity, etc.

ACKNOWLEDGEMENT

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REFERENCES

- [1] S. Ueha, Y. Tomikawa, M. Kurosawa, K. Nakamura, Ultrasonic Motors, Clarendon Press, Oxford, 1993
- [2] T. Sashida, T. Kenjo, An Introduction to Ultrasonic Motors, Clarendon Press, Oxford, NY, 1993
- [3] V.R. Sonti, J.D. Jones, Curved piezoactuator model for active vibration control of cylindrical shells, AIAA Journal 34 (5) (1996) 1034–1040
- [4] W. Soedel, Vibrations of Shells and Plates, Marcel Dekker, New York, 1981
- [5] H.S. Tzou, Piezoelectric Shells (Distributed Sensing and Control of Continua), Kluwer Academic Publishers, Boston/Dordrecht, 1993
- [6] H.R. Shih, Distributed vibration sensing and control of a piezoelectric laminated curved beam, Smart Materials and Structures 9 (6) (2000) 761–766

New Identification Method for Dynamic Characteristics of Roller Guideway Based on Finite Element Model

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INTRODUCTION

Linear guideways are commonly used in feed drives of machine tool, and it can provide highly accurate and stable feed motion. Dynamic stiffness of the machine tool, which dominates the robustness against the cutting force and determines the stability limit of self-excited chatter vibration, is greatly affected by the dynamic characteristics of a linear rolling guideway. The dynamic characteristics of a rolling guideway are defined by the static stiffness and damping which are caused by the contact between the raceway and the rolling elements. However, modeling of the contact parts and identifying its parameters are difficult.

In the previous studies, some researchers have calculated the static stiffness of rolling guideway based on Hertz contact theory and applied it to the theoretical modal analysis of a single carriage of rolling guideway by finite element (FE) analysis [1-5]. Brecher et al. have experimentally identified both the stiffness and damping by comparing the difference of the frequency response functions of the carriage and its dummy. And they have defined the stiffness and damping of rolling guideway by springs which have six degrees of freedom [6]. Although the modal analysis of feed drives is important for designing the machine tool, some of these researches have targeted only at the single carriage. In addition the identification method is complex.

This paper presents a new identification method of stiffness based on the simple experiment and FE analysis. This method gives both the vertical and lateral stiffness of the single carriage of rolling guideway. The number of the parameters which are needed for identifying is small. After the identification, the stiffness is applied for the modal analysis of the single carriage of rolling guideway and the simplified feed drive by FE analysis.

OVERVIEWS OF IDENTIFICATION METHOD

Figure 1 shows the procedure of the identification method for the stiffness of the rolling guideway. It is composed of both the experiment and the FE analysis.

In the experiment part, the frequency response functions (FRFs) of the carriage of rolling guideway are measured by the impulse testing. Since the damping of rolling guideway is low, the resonance frequency is nearly equal to the natural frequency. Thus the resonance frequencies of the 2nd pitching and rolling modes of the carriage are read as the natural frequencies from FRFs. By contrast in the FE analysis part, the vertical and lateral stiffness of rolling guideway are identified using the FE model of the single carriage and the measured natural frequency.

DETAILS OF EXPERIMENTAL PART

Figure 2 shows the experimental setup for measuring the FRFs. The rail of rolling guideway is fixed on the base used for the feed drive. The carriage is inserted to the rail and placed on the center of the rail. The column is attached on the carriage by the bolts. The column has roles to turn down the natural frequency and amplify the angular displacement of the carriage, and makes it possible to obtain acceleration signal with high signal-to-noise ratio.

In this paper, a commercial roller guideway (size: #35, preload: middle, lubrication: VG68 oil) is used for the experiment.

The FRFs of rolling guideway are measured by using impulse testing. The column tip is excited by the impulse hammer in which built the force transducer, and the excitation force is detected. The response acceleration at the column tip is detected by accelerometer which is glued on the

column. The acceleration and force signals are input to the FFT analyzer, and the frequency response functions are calculated. Then the FRFs are output to the personal computer (PC). The excitation is conducted in the lateral and feed directions as shown in Fig. 2.

Figure 3 shows the FRFs of the carriage which was excited in the feed and lateral directions, respectively. The compliance is displayed with logarithmic expression. As shown in the figure, the FRFs have some resonance peaks in each excitation direction. The first two resonance peaks in the feed direction are 1st and 2nd pitching modes of the carriage, and also these in the lateral direction are 1st and 2nd rolling modes. The mode shapes that the rigid rolling and pitching motion of carriage are observed without the

elastic deformation of the column are shown in the 1st modes, respectively. And also the rigid motions of carriage with the elastic deformation of the column are observed in the 2nd mode. These mode shapes are pre-experimentally clarified by the experimental modal analysis.

DETAILS OF FE ANALYSIS PART

In this part, the FE model of the carriage and the column are constructed as shown in Fig 4. The column and the carriage are modeled as the unitary construction with 3D CAD software, and the FE model is created from STEP file of the 3D CAD model by quadratic tetrahedral elements. The FE analysis software used in this paper is MSC. Nastran. The numbers of the tetrahedral elements and nodes in the whole FE model are 31166 and 47851, respectively. In this model, the

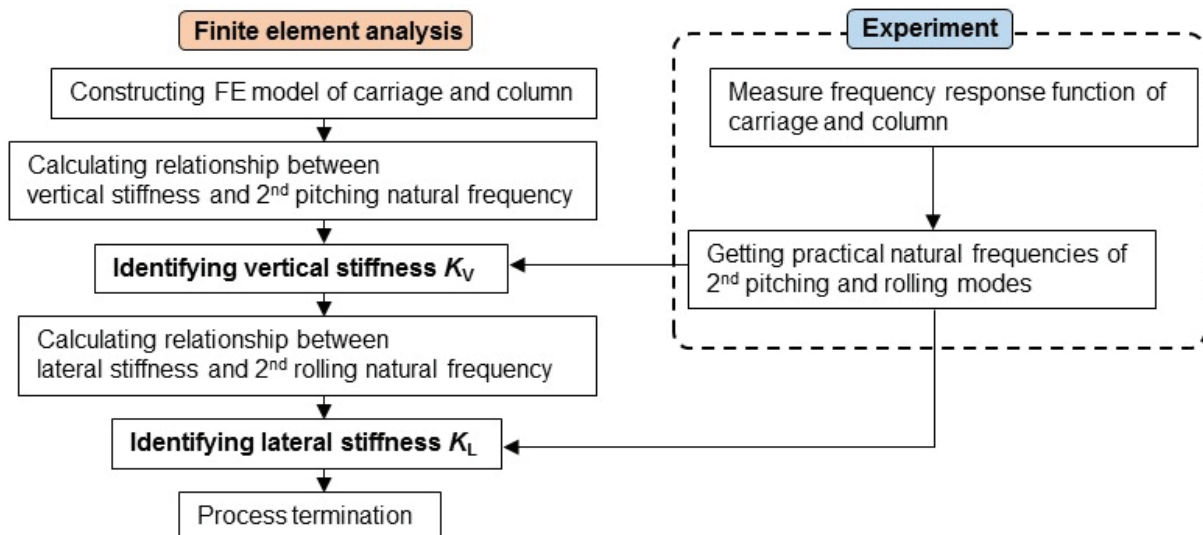


Figure 1. Proposed identification procedure of the vertical and lateral stiffness of rolling guideway

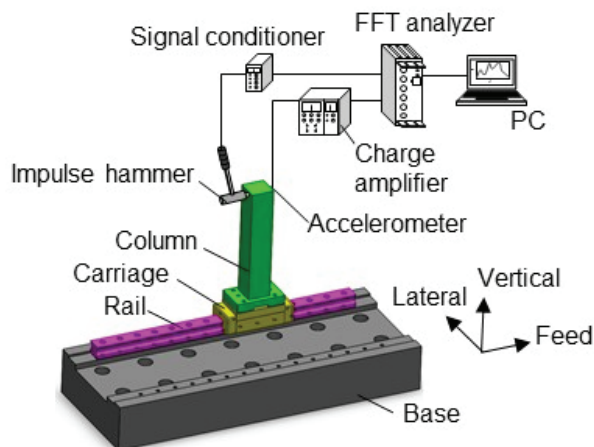


Figure 2. Experimental setup for the identification

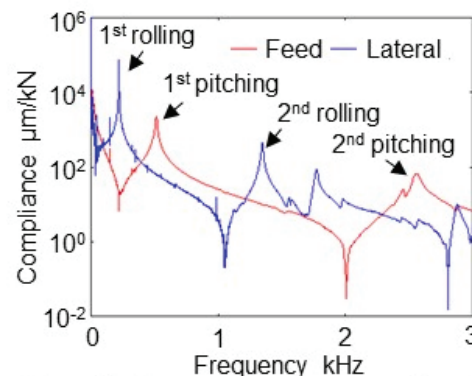
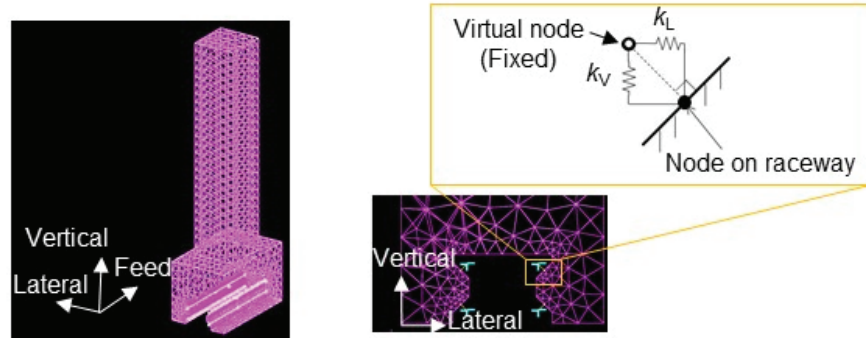


Figure 3. Frequency response functions for identification



(a) Whole finite element model (b) Contact model between raceway and roller

Figure 4. Finite element model of rolling guideway for the stiffness identification

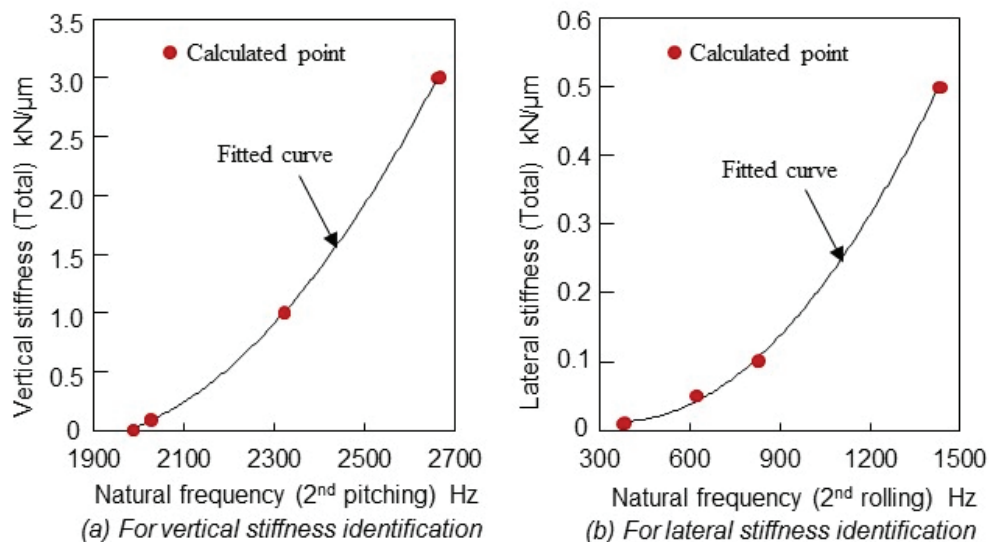


Figure 5. Relationship between the stiffness and the natural frequency

components except the carriage and the column are not considered for simplifying the problem.

The contact parts between the raceway and the rollers are modeled by the linear spring elements as shown in Fig. 4(b). Each fixed node is placed at the points which distance between the fixed node and the node on the raceway is 5 mm in the normal direction of the raceway. This distance is arbitrary. The node on the raceway is placed on at the center of the raceway width. Each pairs of two nodes are connected in the lateral and vertical directions by the spring elements, which are defined in the global coordinate system, respectively. In this paper, 51 pairs of the nodes exist in each raceway, and 408 linear spring elements are used. If the number of the spring elements sets to the number of the rolling elements which contact with the raceway, the stiffness of a single spring element means the

component of the elastic contact stiffness in the vertical and the lateral directions.

In order to calculate the relationship between the natural frequency and the vertical stiffness, modal analysis is conducted by the FE analysis software. In the analysis, Young's modulus of carriage and column are $E = 209$ GPa, Poisson ratio $\nu = 0.3$, density $\rho = 7800$ kg/m³.

Figure 5(a) shows the relationship between the vertical stiffness and the natural frequency of 2nd pitching mode. FE analysis is conducted for four different values of vertical stiffness, and the relationship is obtained by curve fitting. Since the lateral stiffness does not critically affects to the relationship between the vertical stiffness and the natural frequency of 2nd pitching mode, it can be set into arbitrary value. From the relationship shown in Fig. 5(a) and the measured natural

frequency of 2nd pitching mode, the vertical stiffness K_v can be determined as the sum of the stiffness value of the spring elements in the vertical direction.

Figure 5(b) shows the relationship between the lateral stiffness and the natural frequency of the 2nd rolling mode. The calculation of this relationship must be done after calculating the relationship between the vertical stiffness and the natural frequency of 2nd pitching mode because the natural frequencies of rolling modes are affected by both the vertical and the lateral stiffness [3]. As in case of the vertical stiffness, the lateral stiffness is identified from both Fig. 5(b) and the measured natural frequency of the 2nd rolling mode.

As the results, the identified stiffness in the vertical direction is five times larger than that in the lateral direction. In theoretical, the stiffness in both vertical and lateral stiffness of cross roller guideway are same. The stiffness value, however, practically has the difference in each direction because of the elastic deformation of the carriage.

VALIDITY OF IDENTIFIED STIFFNESS

The modal analysis of a single carriage of rolling guideway is conducted for validating the identified stiffness values and the FE model of the contact parts. The resonance frequencies of the single carriage shown in Fig. 6(a), which are obtained by both the impulse testing and the modal analysis, are compared. Although the added mass (4.7 kg) is fixed on the carriage for decreasing the natural frequency of the carriage, it does not affect the mode shapes of the natural vibrations. In the analysis, Young's modulus of carriage and column $E = 209$ GPa, Poisson ratio $\nu = 0.3$, density $\rho = 7800$ kg/m³. The FE model of the feed drive is constructed as the monolithic structure and its bolted joints are ignored. The FE model of the feed drive is constructed by 29217 quadratic tetrahedral elements, 44672 nodes, and 408 linear spring elements.

Figure 6(b) shows the resonance frequencies of a single carriage obtained from both the experiment and FE analysis, and Fig. 7 shows these mode shapes obtained from the FE analysis. In the experiment, the vibration mode is recognized from the results of the experimental modal analysis of the single carriage. Comparing the natural frequency shown in Fig. 7, the results

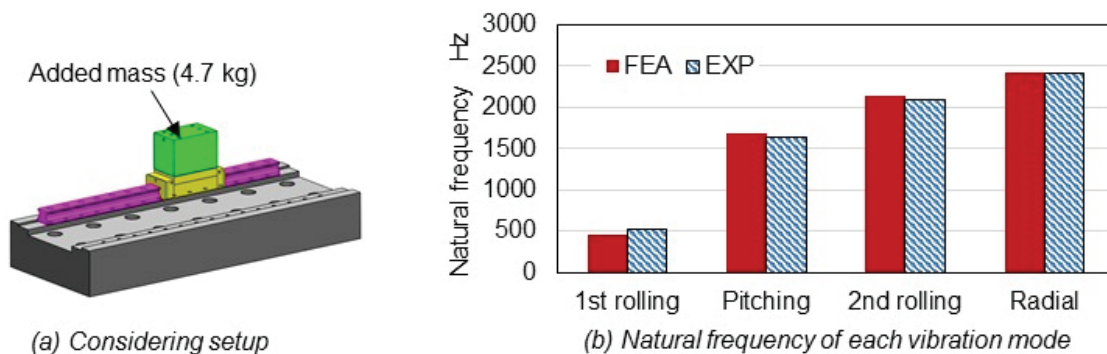


Figure 6. Comparison of the natural frequency of a single carriage of rolling guideway

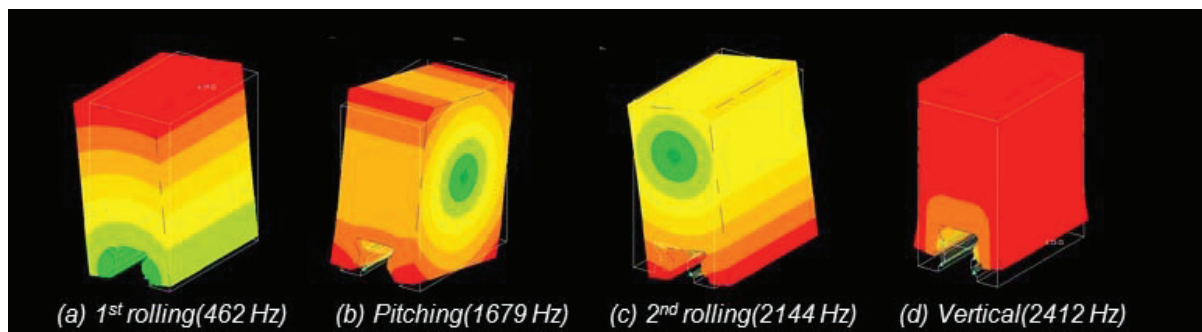


Figure 7. Mode shapes of a single carriage of rolling guideway given by FEA

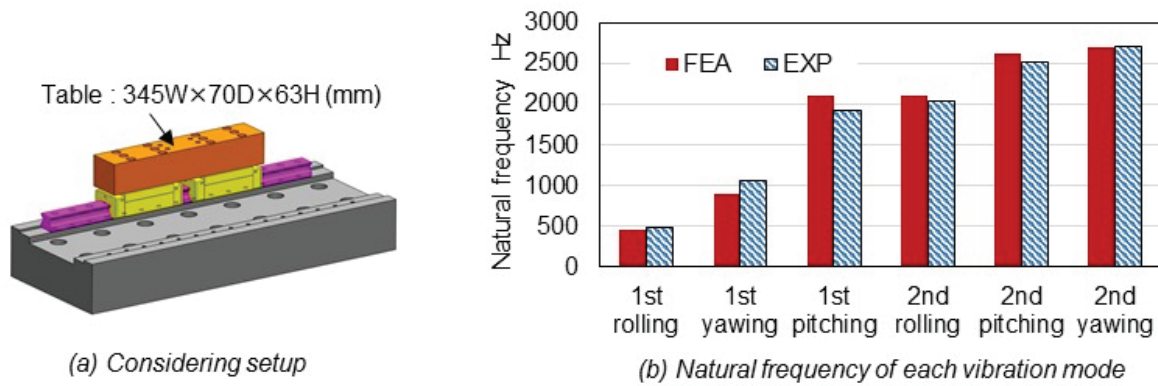


Figure 8. Comparison of the natural frequency of simplified feed drive mechanism

calculated by FE analysis agree rather well with experimental results. Thus the proposed stiffness identification method is useful for the modal analysis of a single carriage of rolling guideway.

MODAL ANALYSIS OF FEED DRIVE

The rolling guideway is not commonly used alone, but is used for the systems such as the feed drive of machine tool. Thus it is important to apply the identified stiffness value and FE model of rolling guideway to the modal analysis of feed drive. However it has not been adequately done in previous researches. In this section, the stiffness which is identified from the single carriage is used for the modal analysis of the feed drive, and the validity of the analysis results is confirmed by comparing with the experiment.

Figure 8(a) shows the simplified feed drive of machine tool and it is constructed by a base, a rail and two carriages, a rectangular bar as a table. The FE model of the feed drive is constructed as the monolithic structure and its bolted joints are ignored. The stiffness of each carriages of rolling guideway is identified by the above mentioned method. The FE model of the feed drive is constructed by 60402 quadratic tetrahedral elements, 92141 nodes, and 816 linear spring elements.

Figure 8(b) shows the resonance frequencies of each vibration mode which is experimentally observed in the frequency response function of the feed drive. The natural frequencies obtained from FE analysis are displayed in the same figure. The calculated results agree rather well with experimental results. Thus the proposed stiffness identification method is useful for the modal analysis of feed drives.

The FE model for identification described in this paper does not include the rail. Although the elastic deformation of the rail affected the stiffness of rolling guideway, the identified stiffness value includes the influence of its elastic deformation. If the rail is in considering for the FE model for identification, the stiffness value of rolling guideway will increase.

CONCLUSION

In this paper, a new stiffness identification method of rolling guideway is proposed. The method consists of two parts, which are an experimental part and a finite element analysis part. It simply gives the vertical and the lateral stiffness of rolling guideway. Stiffness of rolling guideway is the fundamental and important parameter for designing machine tool. Although linear rolling guideway manufacturers can give the stiffness value depend on the case, most of them cannot give it for the modal analysis by FE analysis. Using this new method, the stiffness of rolling guideway can be simply identified and the method gives the value which can be applied for the modal analysis.

Other important parameters for the dynamics of rolling guideway is damping capacity. This method can measure the frequency response function of rolling guideway with high signal to noise ratio. Thus the damping will be evaluated precisely. In the future work, the damping identification of rolling guideway by applying this method will be conducted. It will stimulate the dynamic stiffness analysis of whole machine tool.

In addition, the nonlinear friction in the microscopic displacement region gives the additional stiffness and damping to some natural vibration modes [7]. Thus, these additional stiffness and damping should be considered for

the modal analysis. The authors have been modeled the nonlinear friction characteristics as the complex spring [8]. The complex spring will be introduced to the FE modal analysis.

REFERENCES

- [1] Wu JS, Chang J, Tsai G, Lin C, Ou F. The effect of bending loads on the dynamic behaviors of a rolling guide. *Journal of Mechanical Science and Technology*. 2012;26(3):671–680.
- [2] Hung JP. Load effect on the vibration characteristics of a stage with rolling guides. *Journal of Mechanical and Technology*. 2009; 23: 89–99.
- [3] Ohta H. Sound of linear guideway type recirculating linear ball bearings. *Journal of Tribology*. 1999; 121(4): 678–685.
- [4] Lin CY, Hung JP, Lo TL. Effect of preload of linear guides on dynamic characteristics of a vertical column–spindle system. *International Journal of Machine Tools and Manufacture*. 2010; 50(8): 741–746.
- [5] Mi L, Yin G, Sun M, Wang X. Effects of preloads on joints on dynamic stiffness of a whole machine tool structure. *Journal of Mechanical Science and Technology*. 2012; 26(2): 495–508.
- [6] Brecher C, Fey M, Baumler S. Identification method for damping parameters of roller linear guides. *Production Engineering*. 2012; 6(4-5): 505–512.
- [7] Sakai Y, Tanaka T, Tsutsumi M. Influence of the Excitation Force Amplitude on Vibration Characteristics of a Linear Rolling Bearing. *Journal of the Japan Society for Precision Engineering*. 2014, 80(8): 783–791 (in Japanese).
- [8] Sakai Y, Tanaka T, Yoshioka H, Zhu G, Tsutsumi M, Saito Y. Influence of nonlinear spring behavior of friction on dynamic characteristics of a rolling guideway. *Journal of Advanced Mechanical Design, Systems, and Manufacturing*. 2015, 9(1): 1–15.

DESIGN OF A SUSPENDED MATRIX SENSOR FOR PRESSURE MEASUREMENT

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ABSTRACT

This paper proposes the design and fabrication of a novel sensor system for the measurement of a predesigned target pressure. This sensor system is placed between the testing object and the impact tool. It is featured for transferring the impact pressure of the tool to a rigid or deformable target object. By changing the structural design parameters, the sensor system is set to measure the different pressures. The system is composed of a rigid housing and a suspended sensor matrix. This design is disposable, low-cost and fabricated by means of Fused Deposition Modeling (FDM).

INTRODUCTION

Pressure sensors are widely used in industry for the measurement of the loading on the structure, the status of gas and liquid, etc. Most pressure sensors are used to measure the variation of the pressure. However, some others are designed to measure a target pressure or to be used as a switch.

Much work has been done in the design of pressure sensor. Shi et al. [1] present a 3D printed structural sensor matrix embedded underneath the object for a pressure measurement. However this method is an indirect measurement because the pressure on the surface of the sample is distributed in the object. The pressure measured by the bottom sensor is different from the top impact pressure. Sander et al.[2] designed a monolithic capacitive sensor. Someya et al.[3] designed a flexible pressure sensor matrix for the application of artificial skin. A lot of Micro-electro-mechanical Systems (MEMS) [4, 5] designs are proposed for the motion and pressure measurement[6, 7, 8, 9, 10, 11]. However, the cost of these pressure sensors and structures are high. Additive manufacturing is widely used for rapid fabrication. Based on additive manufacturing, we can build the sensors in a low cost. The most common 3D fabrication of polymer objects is Fused Deposition Modeling (FDM). Some other 3D fabrication methods like selective laser melt-

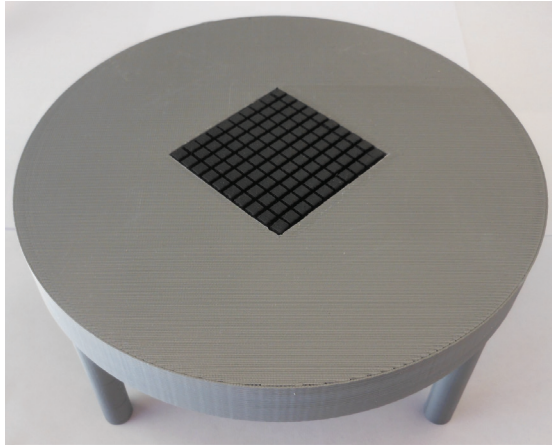
ing (SLM), selective Laser Sintering (SLS), fused filament fabrication (FFF) and stereolithography (SLA) can be used for some other materials or for a higher precision. However, the cost of these methods is higher than FDM.

In this paper, we propose a sensor system to measure a target pressure of a impact tool on a testing object. The pressure sensor is mounted between a impact tool and a testing object to passively measure whether the impact pressure on the testing object reaches a predesigned target pressure. It is featured for the ability of transferring the pressure of the tool to the testing object, which can be rigid or deformable. As shown in Figure 1, we present a sensor matrix with the adoption of the combination of the vertical beam and the narrow flexural horizontal beam [12, 13, 14, 15]. The rest of the paper is organized as follows: We first illustrate the design and the measurement algorithm of the sensor system. Then, we present the fabrication of FDM and the modification of the design. The design and fabrication parameters are discussed and one application of the sensor is also proposed. In the application, the sensor system is mounted on a biosimulant artifact to measure the pressure for a robot impact testing.

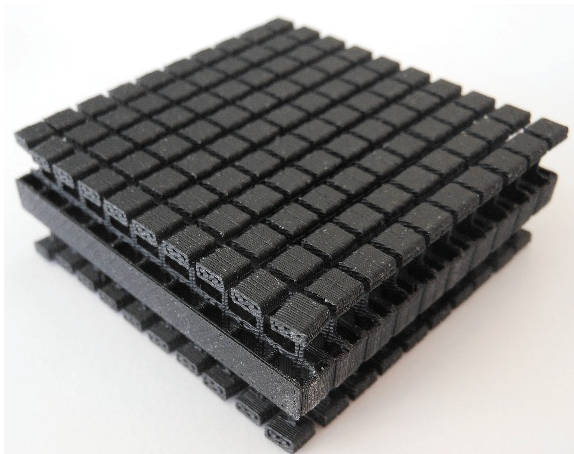
DESIGN OF SENSOR SYSTEM

In this section, we explain the design and measurement algorithm of the sensor system. As shown in Figure 1, the sensor system consists of two parts: a rigid housing and a suspended sensor matrix. The rigid housing is used to suspend the matrix above the testing object. The height of the housing is determined by the sample thickness. The sensor matrix is the layout of the pressure cells. Each pressure cell is a single unit, which reflects the loading contact area.

Figure 2 shows a pressure cell located on the edge of the matrix, so it is suspended directly by the rigid housing. One pressure cell is composed of five parts: top cube, bottom cube, center vertical beam, side horizontal beam, and rigid grid.



(a)



(b)

FIGURE 1. Design of suspended structural pressure sensor.

The top cube of each cell is independently held by one center vertical beam. Depending on the contact surface of the target to be measured, a number of the top cubes are loaded with pressure so that the top cubes will move downwards. The bottom cube is used to contact the sample. The bottom cubes are also independent between each other like the top cubes. The top and bottom cubes are connected by the center vertical beam, which transmits the loading pressure from the top contact surface of the tool to the bottom contact surface of the sample. The vertical beam is designed to be destroyed under a specific target pressure. The top cube will be driven towards to show an obvious sign of buckling. The side

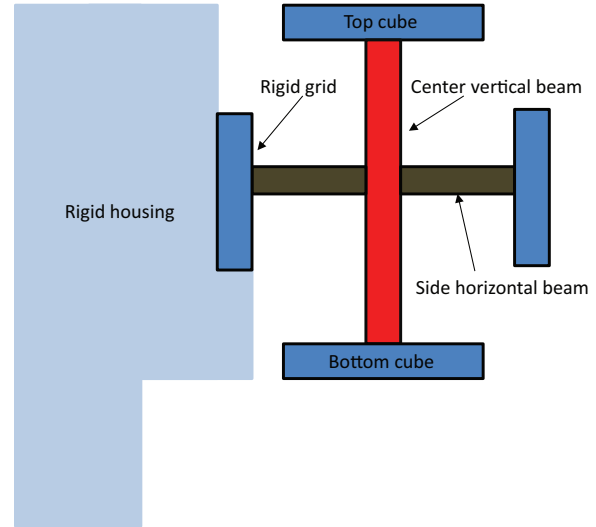


FIGURE 2. Schematic drawing of the sensor system.

horizontal beam of the pressure cell is used to hold the center vertical beam within a small deformation. The horizontal beam also suspends the vertical beam on the rigid grid to ensure the top cubes are laid in a plane before testing. The horizontal beam breaks earlier than the vertical beam. The grid here is used to isolate each pressure cell to make each pressure cell is independently affected by the loading force. This setting enables the structure to measure the pressure instead of the force.

Figure 3 illustrates the measurement algorithm of the sensor system. The steps of the measurement are shown as below:

1. Before testing, the impact tool is located above the sensor system and the testing object is underneath the sensor matrix. The top surface of the testing object is attached to the bottom surface of the sensor matrix without any pressure.
2. The impact tool once hits the top cubes of the sensor matrix. A number of the cubes contacted is corresponding to the contact area of the tool. Thus, the total loading force of the tool is transmitted to the forces loaded on the top cubes, which cause the downward movement of the cube.
3. The top cube is attached to the center vertical beam and the vertical beam is suspended by the side horizontal beam on the grid. At this step, the impact force on the top cube

is transmitted to the testing object. The testing object is deformed by the bottom cubes, of which the quantity is the same as the top moving cubes. The suspended side beam is deformed largely, while the vertical beam has small deformation along the axial direction. According to the types of the suspended beams, they are treated as a cantilever beam or a beam with two fixed ends.

4. In this step, the top cube, vertical beam and the bottom cube keep moving downward and deform the testing object. When the horizontal beam is bent too much and the local stress reaches the maximum ultimate strength, the horizontal beam will be broken.
5. Since the vertical beam is suspended by the side beam to the grid, the vertical beam is free from the grid after the horizontal side beam is broken. The vertical beam keeps deforming in the axial direction due to the loading on the top cube along the vertical beam. At a certain position, the vertical beam will buckle when the axial loading reach the maximum allowed force.
6. After the vertical beam buckles, the vertical beam can show a clear sign to be examined when we take the sensor matrix out.

FUSED DEPOSITION MODELING AND MODIFICATION

In this section, we illustrate the fabrication and the modification of the sensor matrix. The fabricated structural sensor matrix is shown in Figure 4. Figure 4 (a) is the top view of the matrix. We can find that the top cubes are independent from each other. These top cubes are well fabricated in a uniform shape. The gaps between each two top cubes are clear and there is no extra plastic filament. Figure 4 (b) is the side view of the matrix. The long vertical beams are fabricated to support the top cubes and the bottom ones. The dimensions of the vertical beam determine the maximum allowed loading. For example, when the beam length is longer and the cross section is smaller, the maximum allowed loading value is lower.

As the side beam needs to be broken before the vertical beam buckles, we present two types of design as shown in Figure 5. The side beam in Figure 5 (a) is a suspended beam with two fixed

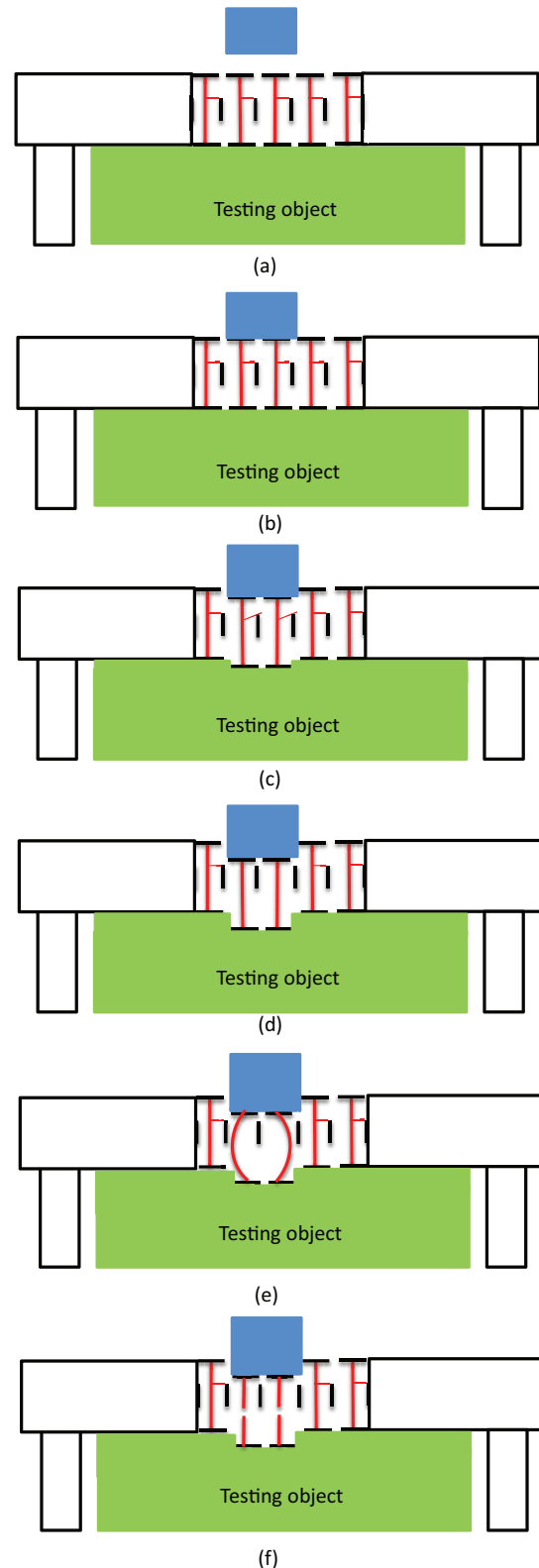


FIGURE 3. Measurement algorithm of the sensor system.

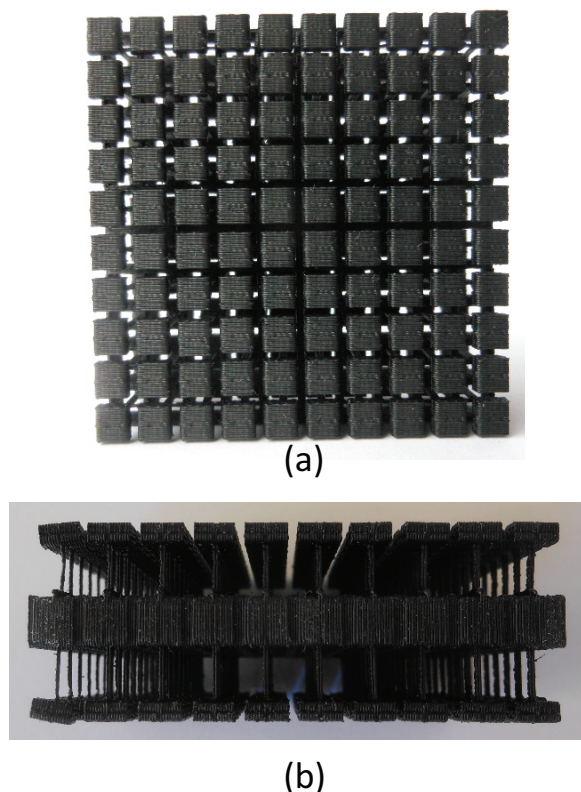


FIGURE 4. Sensor matrix and single pressure cell.

ends on the grid. This design improves the fabrication survivability of the structure, however, this horizontal beam is not easy to deform and break in the steps of Figure 3 (c) and (d). Figure 5 (b) shows a half horizontal beam. This beam structure is a cantilever beam so it is much easier to deform. However, this beam is not a symmetrical beam. The loading force on the top cube may cause the vertical beam to be tilted.

In order to print the matrix of the pressure cell, supporting material is used in the printing process shown in Figure 6. Here, we use FDM in the fabrication of the sensor matrix. Figure 6 (a) shows that the vertical beam and the horizontal beam are in the same plane, because the nozzle of the printer moves continuously in a plane. From Figure 6, we find that the supporting material surrounds the main printing material. The structural parameters of the design are constrained by the capabilities of the 3D printing method. With a different type of 3D printing technology, the pressure cell can be built smaller, finer feature, or from different materials. By means of 3D printing with higher resolution, we could reduce the size of top

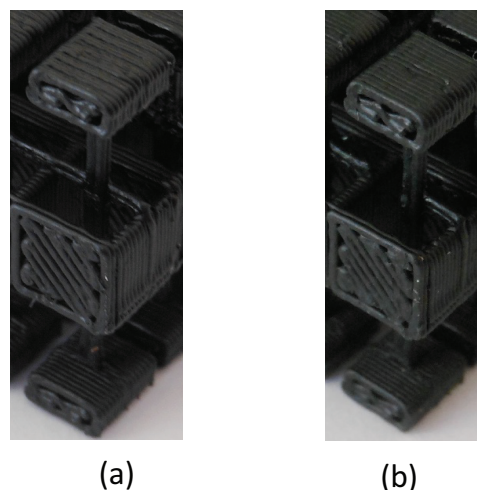


FIGURE 5. Two types of suspended side beams.

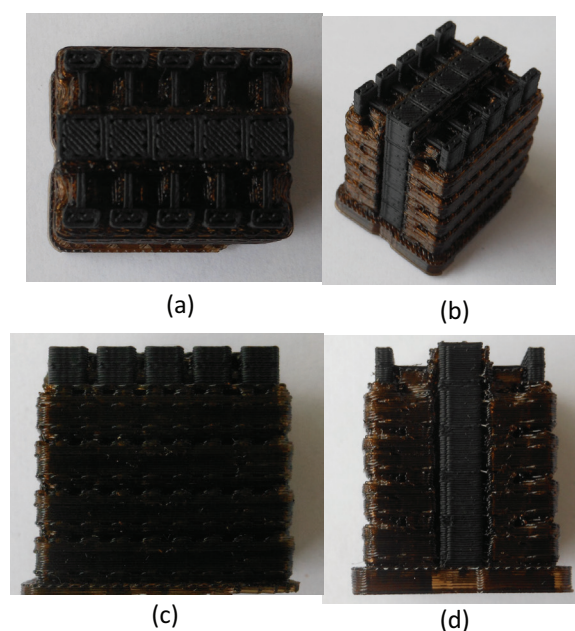


FIGURE 6. Fused Deposition Modeling.

cubes to increase the sensitivity regarding to the size of the object to be measured. For example, SLA commonly has a higher resolution and uses cured material like resin.

DESIGN PARAMETERS

The schematic drawing of the sensor is shown in Figure 7. The cross section of the vertical beam is $0.8\text{mm} \times 0.8\text{mm}$. The cross section of the horizontal beam is $0.4\text{mm} \times 0.4\text{mm}$. The cross section of the vertical beam is larger than that of the horizontal beam, because the horizontal beam is designed to be deformed easier than the vertical beam. w denotes the size of the top and bottom

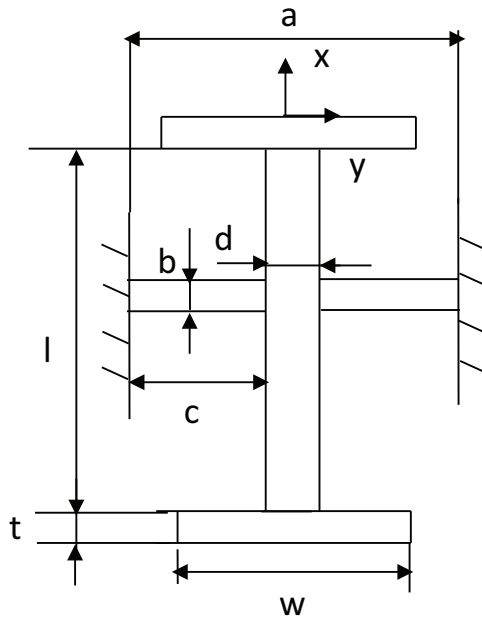


FIGURE 7. Dimension of a pressure cell.

cubes, which is 4mm in this paper. When w is larger, the loading on the pressure is more stable. Otherwise, the vertical loading force may cause the vertical beam bend instead of buckle. a is the dimension of each pressure cell, which is corresponding to the sensitivity of the measurement. a should be as smaller as as possible, because the same contact area on the sensor matrix will cause more independent top cubes to move down. l and d here is corresponding to the target pressure. Because d has the limitation of the fabrication, we can adjust the dimension of l . For example, when we set a higher pressure, we can increase d and reduce the l . When we set a lower pressure, we can reduce d and increase l . When d is reduced the minimum allowed fabrication dimension, we can only increase l .

ONE APPLICATION OF THE SENSOR

Here, we demonstrate the sensor system by means of an application example. The sensor system combines the testing impact tool, testing object sample to build the human robot safety impact testing. In this example, the sensor system is used to measure whether the impact pressure produced by the tool reaches the target pressure on the testing sample.

The testing sample called biosimulant artifact [16], includes a top artificial leather and bottom ballistic gelatin. The top leather simulates human skin and the bottom gelatin simulates human

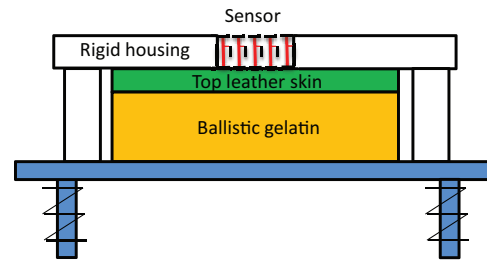


FIGURE 8. Application of robot impact testing.

muscle. The biosimulant artifact is used to simulate the stress distribution when the impact force is applied on the top of the skin. As shown in Figure 8, the sensor system is mounted above the biosimulant artifact. The length of the rods of the rigid housing is adjusted to the thickness of the artifact, so the bottom cubes of the sensor matrix contact the top surface of the leather. As shown in Fig. 8, the sensor system and the biosimulant artifact are set on the stage of the Dynamic Impact Testing and Calibration Instrument (DITCI) [16]. It is a simple instrument, with a significant data collection and analysis capability that is used for the testing and calibration of the pressure sensor system.

During the testing, the impact tool drops from a height. When the tool hits the sensor matrix, a mount of top cubes are contacted to transfer the loading pressure. After the side beams are destroyed, the center vertical beam and bottom cubes will keep deforming the artifact until the vertical beam reaches the maximum allowed stress of buckling. Through the measurements of the calibrated sensor matrix, we could examine whether the impact pressure reaches the pre-designed target pressure.

CONCLUSIONS

In this paper, we present a novel design of a sensor matrix system for pressure measurement. This sensor adopts 3D printing fabrication to build a small size structure. The structure is based on the combination of the supporting horizontal beam and the center vertical beam. Some meaningful conclusions can be drawn as following. (1) The design is used to measure a target pressure by changing the dimension of the vertical beam. (2) The sensor is placed between impact tool and the testing object and it can transfer the pressure of the tool to the testing object. (3) By adjusting the dimensions of vertical beam, we set the pre-designed target pressure. (4) The measurement of the pressure is also decided by the size of the

top cube. However, a larger dimension of the top surface will also result in a lower sensitivity of the measurement. (5) The structural pressure sensor is disposable, low cost, and easy to fabricate. (6) An application example of the sensor is proposed to measure whether the impact pressure of the tool on the biosimulant artifact reaches the target pressure for robot safety testing.

REFERENCES

- [1] Shi H, Kim Y, Dagalakis N, Duan X. A 3D printing flexure pressure sensor for robot impact safety testing. In: The Fifth Asian International Symposium on Mechatronics (AISM). Guilin, China; 2015. Forthcoming.
- [2] Sander CS, Knutti JW, Meindl JD. A monolithic capacitive pressure sensor with pulse-period output. *Electron Devices, IEEE Transactions on*. 1980 May;27(5):927–930.
- [3] Someya STISKYKH T, Sakurai T. A large-area, flexible pressure sensor matrix with organic field-effect transistors for artificial skin applications. *PNAS*. 2004;101(27):9966–9970.
- [4] Shi H, Duan X, Su HJ. Optimization of the workspace of a MEMS hexapod nanopositioner using an adaptive genetic algorithm. In: *Robotics and Automation (ICRA)*, 2014 IEEE International Conference on. Hong Kong, China; 2014, May 31–June 7. p. 4043–4048.
- [5] Shi H, Su HJ, Dagalakis N. A stiffness model for control and analysis of a MEMS hexapod nanopositioner. *Mechanism and Machine Theory*. 2014;80:246 – 264.
- [6] Abeysinghe DC, DasGupta S, Boyd JT, Jackson HE. A novel MEMS pressure sensor fabricated on an optical fiber. *Photonics Technology Letters, IEEE*. 2001 Sept;13(9):993–995.
- [7] Shi H, Su HJ, Dagalakis N, Kramar JA. Kinematic modeling and calibration of a flexure based hexapod nanopositioner. *Precision Engineering*. 2013;37(1):117 – 128.
- [8] Shi H, Su HJ. Workspace of a Flexure Hexapod Nanopositioner. In: *Proceedings of ASME IDETC/CIE*. Chicago, Illinois, USA; 2012, August 12–15. p. 341–349.
- [9] Palasagaram JN, Ramadoss R. MEMS-Capacitive Pressure Sensor Fabricated Using Printed-Circuit-Processing Techniques. *Sensors Journal, IEEE*. 2006 Dec;6(6):1374–1375.
- [10] Krondorfer RH, Kim YK. Packaging Effect on MEMS Pressure Sensor Performance. *Components and Packaging Technologies, IEEE Transactions on*. 2007 June;30(2):285–293.
- [11] Nakamura K ITYTOK Tomohiro. MEMS Pressure Sensor. US Patent. June;8516905 B2.
- [12] Su HJ, Shi H, Yu J. A Symbolic Formulation for Analytical Compliance Analysis and Synthesis of Flexure Mechanisms. *ASME Journal of Mechanical Design*. 2012;134(5):051009.
- [13] Su HJ, Shi H, Yu J. Analytical Compliance Analysis and Synthesis of Flexure Mechanisms. In: *Proceedings of ASME IDETC/CIE*. Washington, DC, USA; 2011, August 28–31. p. 169–180.
- [14] Shi H. Modeling and Analysis of Compliant Mechanisms for Designing Nanopositioners. The Ohio State University. Columbus, OH, USA; 2013.
- [15] Shi H, Su HJ. An Analytical Model for Calculating the Workspace of a Flexure Hexapod Nanopositioner. *ASME Journal of Mechanisms and Robotics*. 2013;5(4):041009.
- [16] Dagalakis NG, Yoo JM, Oeste T. Human-Robot Collaboration Dynamic Impact Testing and Calibration Instrument for Disposable Robot Safety Artifacts;.

BIOSIMULANT ARTIFACT WITH EMBEDDED CALCIUM ALGinate BEAD SENSOR FOR ROBOT IMPACT SAFETY TESTING

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ABSTRACT

This paper presents the design of a disposable biosimulant human tissue artifact system for robot safety testing. It is used to show a clear sign of the severe injuries caused in the case of a robot impact with a human. The fabrication method is described including the design and fabrication of the calcium alginate bead and the embedding procedure of the beads into the biosimulant artifact. The artifact system is tested with a Dynamic Impact Testing and Calibration Instrument (DITCI) from the National Institute of Standards and Technology (NIST). The design is useful for the preparation of new robot safety standards.

INTRODUCTION

The movement of manufacturing to countries featuring labor with low hourly wages over the last fifteen years has motivated the development of a new generation of industrial robots that can work side-by-side with human workers [1]. This has created a new technology of Human-Collaboration-Robotics (HCR), which combines the intelligence and dexterity of humans with the strength, repeatability and endurance of industrial robots [2]. Since most robots are powerful moving machines, the safety of workers working around these robots has become a top priority for safety standards development. We are using biosimulant materials for the fabrication of inexpensive, disposable HCR safety testing artifacts. These testing artifacts will make possible the measurement of forces, pressure and strain when humans and robots come into contact and also the magnitude of injuries caused by robot static and impact pressure. The Dynamic Impact Testing and Calibration Instrument (DITCI) is a simple instrument shown in Figure 1, with a significant data collection and analysis capability that is used for the testing and calibration of biosimulant human tissue artifacts [3, 4].

Various research groups have used human subjects to collect data on pain induced by the clamping force, pressure and maximum impact force of

the HCRs [5, 6, 7]. Although the results of these tests are hard to reproduce and can vary even among subjects of similar characteristics, they can be very useful for the preparation of safety testing standards. Unfortunately, human safety testing is not an option for HCR industrial applications every time there is a change of a tool or control program, so the use of a biosimulant artifact system is expected to be a good alternative.

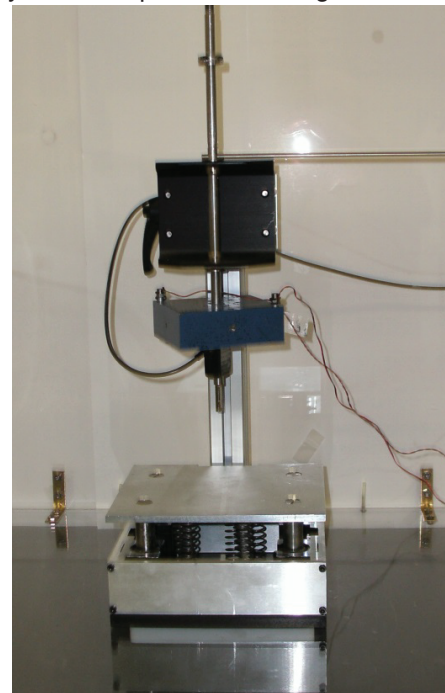


FIGURE 1. Impact testing set up.

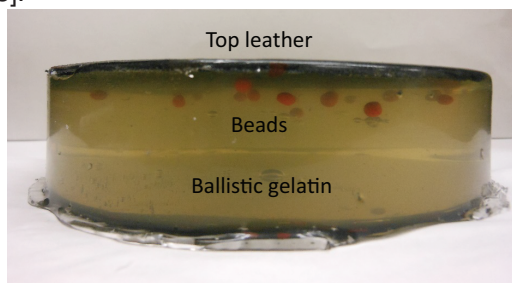
Much work has been done in the design of pressure sensor. Sander et al. [8] designed a monolithic capacitive sensor. Someya et al. [9] designed a flexible pressure sensor matrix for the application of artificial skin. A lot of Micro-electro-mechanical Systems (MEMS) designs are proposed for pressure sensors [10, 11, 12, 13]. However, the cost of these pressure sensors are high. Based on a chemical fabrication method, we could build cheap and disposable measurement system. Daly and Knorr [14] proposed the

fabrication of a chitosan alginate capsule. Hugué and Dellacherie [15] described the fabrication of calcium alginate beads coated with chitosan.

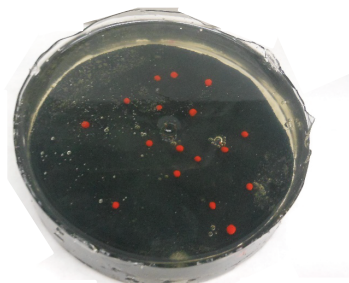
In this paper, we present a chemical based fabrication procedure for a disposable human artifact embedded with calcium alginate bead for robot impact safety testing. The rest of the paper is organized as follows: We firstly present the design methodology of the artifact system. Secondly, the fabrication method of the calcium alginate bead is proposed. Furthermore, the embedding procedure of the bead into human tissue artifact is described. Finally, the biosimulant artifact system is mounted on the DITCI for an impact testing.

DESIGN OF BIOSIMULANT ARTIFACT WITH EMBEDDED BEAD

In this section, we describe the design of the biosimulant artifact system. As shown in Figure 2, the sensor system consists three parts: top leather, soft tissue, and embedded sensor. The top leather is a piece of artificial skin of disk shape. Soft tissue is made of ballistic gelatin. The embedded sensor is a calcium alginate based bead design. The combination of top leather and ballistic gelatin is called the biosimulant artifact [3].



(a) Side view of the artifact with imbedded bead sensors



(b) Bottom view of the biosimulant artifact system

FIGURE 2. Biosimulant artifact and embedded sensor.

The biosimulant artifact simulates human skin and muscle, and simulates the stress distribution when the impact force is applied on the top surface of the skin. The deformation of the ballistic gelatin caused by the dynamic impact force results in the stress distributed on the bead sensors. Thus, the embedded bead sensors will deform corresponding to the deformation of the ballistic gelatin. When the red beads over deform, they will be destroyed above a certain pressure caused by the impact force. The artifact embedded with beads is disposable after testing due to the low fabrication cost.

FABRICATION OF CALCIUM ALGinate BEAD

In this section, we present the fabrication of the calcium alginate bead. The main chemical reaction is based on the mixing of a solution of sodium alginate and calcium chloride. As shown in Figure 3, there are two solutions: base and bath. The base solution is used to drop into the bath solution. Here, the base solution is sodium alginate and the bath solution is calcium chloride.

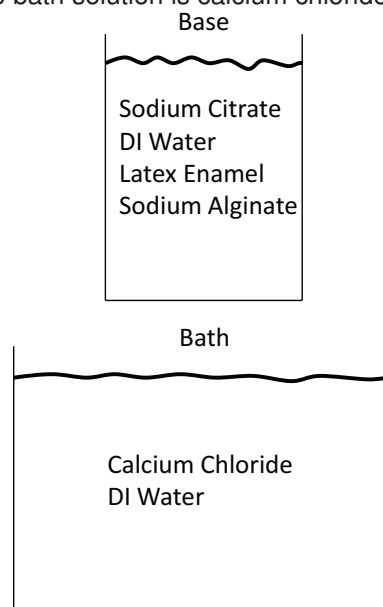


FIGURE 3. Preparation of the base and bath.

The base solution consists of sodium citrate, deionized (DI) water, latex and sodium alginate. The chemicals used in the fabrication are shown in Table 1. Firstly, sodium citrate is dissolved in the DI water to obtain a transparent solution. The sodium citrate is used to remove the calcium ions which will otherwise react with the sodium alginate. Then, red latex enamel is added into the solution. By stirring with a steel stick, we can obtain a red solution without the calcium ion. Finally,

we add sodium alginate. Keep stirring the sodium alginate solution until the sodium alginate powder is fully dissolved. It takes approximately 15 minutes. The bath solution is obtained by dropping calcium chloride into DI water. In order to maintain an easy fabrication, the material used here are mainly based on food recipes. Sodium citrate, sodium alginate, and calcium chloride are purchased from modenist pantry ¹. The latex enamel of gloss cherry is from Valspar.

As shown in Figure 4, the base is dropped into the bath with a pipette. The viscosity of the base should be higher than the one of the bath so that the drop of the base could maintain the original shape of the beads instead of dispersing. If the viscosity is not higher enough, xanthan gum could be added to the base to make the solution stickier.

The pipette is kept with at enough height shown in Figure 4 to ensure the drop can maintain its shape when it contacts the surface of the bath solution. We can continue dropping several drops, but each drop needs to be kept at some distance from the rest so that they do not stick to each other. The beads are kept in the bath for 2 minutes. In this step, the sodium alginate in the base attracts the calcium chloride in the bath. This reaction creates one layer of calcium alginate as shown in Figure 4. The calcium alginate is a kind of membrane which surrounds the inside liquid sodium alginate. After we take out the beads from the bath and soak them in the DI water, we manually stir the water for about 2 minutes to clean the remaining calcium chloride on the outside surface of the membrane. In this step, if the beads are kept over 30 minutes in the DI water, the size of the beads will increase because the calcium alginate membrane is porous and the DI water is absorbed by the beads.

Finally, we take the beads out of the DI water. Instead of drying the beads, we keep them wet and store in the refrigerator. The extra calcium ions in the outside membrane will keep reacting with the inside liquid sodium alginate. After about 12 hours, the bead turns to be a uniform ball, which contains water like a sponge.

¹Certain commercial materials, equipment, and instruments are identified in this paper in order to specify the experimental procedure adequately. Such identification is not intended to imply recommendation or endorsement by the National Institute of Standards and Technology, nor is it intended to imply that the materials or equipment identified are necessarily the best available for the purpose.

TABLE 1. Chemicals of sodium alginate bead.

Chemical	Quantity (g)	Testing (g)
Sodium citrate	0.1	0.15
DI water	10	10.25
Latex enamel	2	2
Sodium alginate	0.08	0.08
Calcium chloride	0.2	0.2
DI water	20	20.6

FABRICATION OF BIOSIMULANT ARTIFACT AND EMBEDDING PROCEDURE OF BEAD

Our current artifact consists of a disk of biosimulant skin and the soft tissue shown in Figure 2. In nature, human skin consists of a thin outer layer called the epidermis, a thick inner layer called the dermis, and subcutaneous fat. The skin is the first layer of defense against impact injuries, which have been studied by medical professionals and forensic researchers. Because cadavers are expensive and difficult to maintain, forensic researchers have been searching for biosimulant materials with mechanical properties similar to those of human skin [16]. There are a few suppliers of this type of material. For all of our artifact test samples, we used chromed tanned cowhide Full Cowhide Side, Upholstery or Garment Leather Black, thickness about 1.1 mm [17].

Water solutions containing 10 % to 30 % mass of gelatin have been studied extensively and are considered to be good human muscle tissue biosimulants [18]. Here are the most important conclusions relevant to our work. For non-penetrating injuries, a water solution containing 20 % mass of gelatin gives a better representation of muscle tissue impact response [19]. For penetrating injuries, a water solution containing 10 % mass of gelatin is more appropriate [20]. Distilled water should be used for the gelatin solutions to avoid contaminants and acidity variations [21]. For all of our test artifact preparations, we used a distilled water solution containing 10% mass gelatin powder supplied by Vyse Professional Grade Ballistic & Ordnance Gelatin [22].

As shown in Figure 5, the solution is poured on top of the skin-biosimulant in an aluminum or glass mold. The glass mold produced artifacts of excellent transparency, and are preferred over the metal molds. The gelatin forms a strong bond with the backside of the skin biosimulant and is observed never to delaminate during impact tests.

After pouring the first layer of ballistic gelatin, we

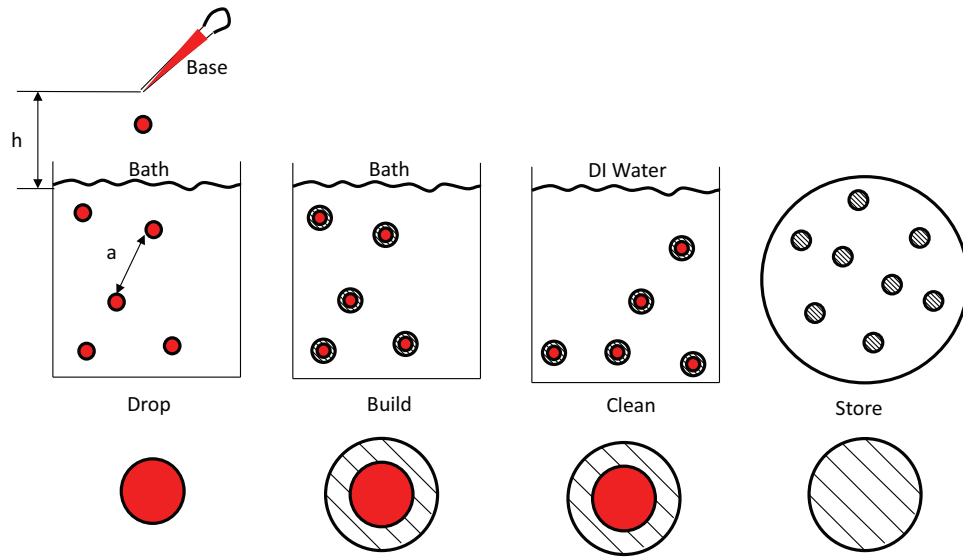


FIGURE 4. Fabrication of the calcium alginate beads.

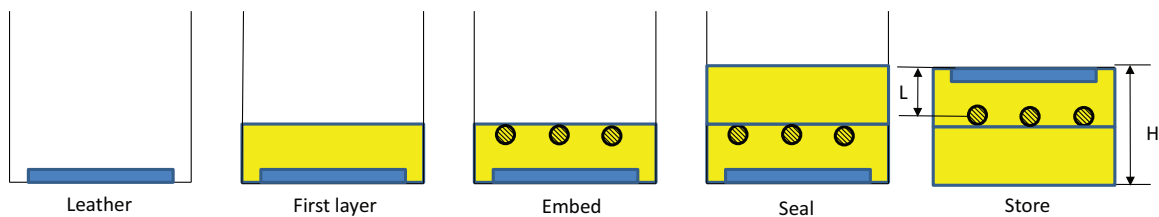


FIGURE 5. Embedding procedure of the beads into the biosimulant artifact.

drop the beads into the liquid gelatin. The beads will float due to the light weight. In order to seal the beads, the liquid gelatin need to be cured. The curing process could be accelerated by placing the gelatin in the refrigerator. When the first layer of the gelatin is partially cured, we pour the liquid gelatin for the second layer. Once the second layer gelatin is fully cured, the embedding procedure is completed. The most important thing in embedding is the temperature control of the first and second layer gelatin. The first layer gelatin needs to be cool enough in order to seal the beads. The temperature of the poured liquid gelatin for the second layer needs to be around 30 degree Celsius. If the temperature is too high, the gelatin will heat up the contact surface of the first layer gelatin. This results in releasing the sealed beads, which will float to the top surface of the second layer gelatin. If the temperature is too low, the bonding of the first and second layer will be not strong enough to remain bonded throughout the impact testing.

As shown in Figure 5, we take out the artifact system from the mold after the ballistic gelatin is fully cured. The height of the artifact corresponds to the thickness of human tissue. In this paper, the height of the used artifact was 5 cm (4") inches because it simulates the soft tissue of human belly. The beads need to be sealed in the gelatin because they are porous and contain water. If the beads are dried, they turn to behavior like rubber.

IMPACT TESTING

In this section, we demonstrate the artifact system in the robot impact testing, shown in Figure 1. The fabrication parameters of the sample bead are shown in Table 1. The tool attached to the weight drops on the artifact. This simulates the impact on the human belly.

As shown in Figure 6, the tool hits the leather first, and then deforms the gelatin and the embedded red beads. Here, we use a tool with a cross section of 5 mm × 6 mm. The force sen-

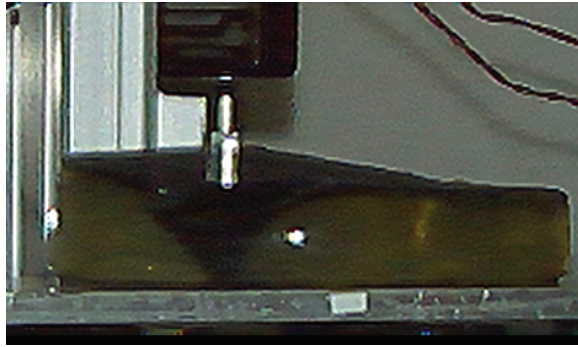


FIGURE 6. *Impact testing of bead sensor embedded artifact system.*

sensor fixed on the tool records a maximum value of 118 *N*. Thus, the corresponding pressure is 3.93 *N/mm*². This means that the tool has hit the organism under the belly. Figure 7 (a) shows the original embedded bead sensor. Figure 7 (b) shows the destroyed bead after the impact testing.

After we section the ballistic gelatin, we can see the destroyed bead sensor in Figure 7 (c).

CONCLUSION

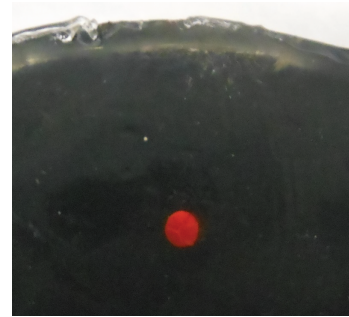
In this paper, a design of disposable biosimulant human tissue artifact system is proposed. It is used for the measurement of the injuries caused by a robot impact force. The fabrication and embedding procedure of calcium alginate bead sensors are described. In the impact testing, the bead is destroyed and shows a clear sign that the tool hit the artifact severely. This design can be used in the preparation of artifact safety test standards.

ACKNOWLEDGMENT

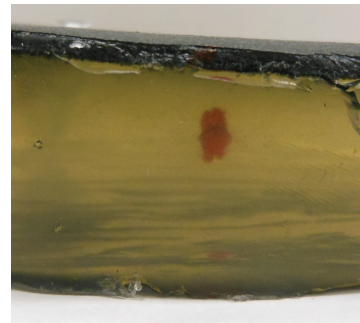
This work was supported by the Robotic Systems for Smart Manufacturing Program of the Intelligent Systems Division, Engineering Laboratory, National Institute of Standards and Technology, USA.

REFERENCES

- [1] Guizzo E, Ackerman E. How Rethink Robotics Built Its New Baxter Robot Worker, *IEEE Spectrum*; 2012.
- [2] Pine A. Just Ahead: The Robotics Revolution; Jan 8, 2013.
- [3] Dagalakis NG, Yoo JM, Oeste T. Human-Robot Collaboration Dynamic Impact Test-



(a) Before impact testing



(b) After impact testing



(c) The destroyed sensor

FIGURE 7. *Impact testing results of the sensor system.*

ing and Calibration Instrument for Disposable Robot Safety Artifacts. *Industrial Robot: An International Journal*; Accepted.

- [4] Shi H, Kim Y, Dagalakis N, Duan X. A 3D printing flexure pressure sensor for robot impact safety testing. In: *The Fifth Asian International Symposium on Mechatronics (AISM)*. Guilin, China; 2015. Forthcoming.
- [5] Haddadin S, Albu-Schaffer A, De Luca A, Hirzinger G. Collision detection and reaction: A contribution to safe physical Human-Robot Interaction. In: *Intelligent Robots and Systems, 2008. IROS 2008. IEEE/RSJ International Conference on*; 2008. p. 3356–3363.

- [6] Haddadin S, Albu-Schaffer A, De Luca A, Hirzinger G. Evaluation of Collision Detection and Reaction for a Human-Friendly Robot on Biological Tissues. IARP International Workshop on Technical challenges and for dependable robots in Human environments 2008;.
- [7] (DLR) GAC. Safe Human-Robot Interaktion;Available from: [http:// www. youtube. com/ watch?v= dnUwqngH0bM](http://www.youtube.com/watch?v=dnUwqngH0bM).
- [8] Sander CS, Knutti JW, Meindl JD. A monolithic capacitive pressure sensor with pulse-period output. *Electron Devices*, IEEE Transactions on. 1980 May;27(5):927–930.
- [9] Someya STISKYKH T, Sakurai T. A large-area, flexible pressure sensor matrix with organic field-effect transistors for artificial skin applications. *PNAS*. 2004;101(27):9966–9970.
- [10] Abeysinghe DC, Dasgupta S, Boyd JT, Jackson HE. A novel MEMS pressure sensor fabricated on an optical fiber. *Photonics Technology Letters*, IEEE. 2001 Sept;13(9):993–995.
- [11] Palasagaram JN, Ramadoss R. MEMS-Capacitive Pressure Sensor Fabricated Using Printed-Circuit-Processing Techniques. *Sensors Journal*, IEEE. 2006 Dec;6(6):1374–1375.
- [12] Krondorfer RH, Kim YK. Packaging Effect on MEMS Pressure Sensor Performance. *Components and Packaging Technologies*, IEEE Transactions on. 2007 June;30(2):285–293.
- [13] Nakamura K ITYT, K O. MEMS Pressure Sensor. US Patent 8516905 B2;.
- [14] Daly MM, Knorr D. Chitosan-Alginate Complex Coacervate Capsules: Effects of Calcium Chloride, Plasticizers, and Polyelectrolytes on Mechanical Stability. *Biotechnology Progress*. 1988;4(2).
- [15] Huguet ML, Dellacherie E. Calcium Alginate Beads Coated with Chitosan: Effect of the Structure of Encapsulated Materials on Their Release. *Process Biochemistry*. 1996;31(8):745–51.
- [16] Jussilaa LAPM J, Kulomaki E. Ballistic skin simulant. *Forensic Science International*. 2005 May;150:63–71.
- [17] Brettuns Village Leather M Lewiston;Available from: [http://www. brettunsvillage. com/ leather/ sides.htm](http://www.brettunsvillage.com/leather/sides.htm).
- [18] Harvey MJH E N, Butler EGea. Mechanism of wounding. in: J.B. Coates (Ed.), *Wound Ballistics*. US Army Surgeon General, Washington DC. 1962;p. 143–235.
- [19] Bir C. The evaluation of blunt ballistic impacts of the thorax. Wayne State University. Detroit, Michigan,USA; 2000.
- [20] Viano DC, King AI. Biomechanics of chest and abdomen impact. CRC Press, Boca Raton; 2008.
- [21] Jussilaa J. Preparing ballistic gelatin-review and proposal for a standard method. *Forensic Sci Int*. 2004 May;141:91–98.
- [22] Gelatin Innovations I Schiller Park;Available from: <http://www.gelatininnovations.com/>.

HIGH RESOLUTION FLEXURE-BASED TIP-TILT TABLE

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INTRODUCTION

To achieve nanometer-level error motion for precision air bearing spindles, metrology feedback of the individual spindle components is important. The setup shown in Figure 1 illustrates the measurement of a spindle shaft on a roundness tester with four indicators. This measurement provides roundness, circular flatness, parallelism and squareness in a single setup. These measurement results are used as process feedback during manufacture.

A key feature of the roundness tester that enables the measurement of squareness is the tip-tilt table. This paper describes the design and testing of a flexure based tip-tilt table suitable for use in roundness testers and machine tools with 0.2 arc second resolution.

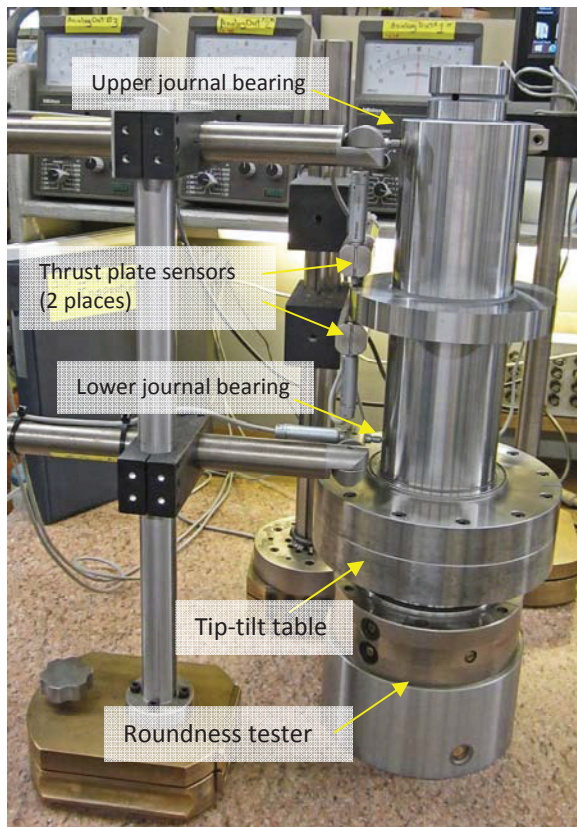


FIGURE 1. Measuring a shaft for roundness, flatness, parallelism and squareness in a roundness tester with four sensors.

MOTIVATION

Air bearings benefit from an averaging affect in which radial error motions can be many times less than the out-of-roundness of the stator and rotor. However, for lower frequency harmonic errors, such as fundamental axial error, the averaging effect is less pronounced. As a result, we have developed a specialized measuring setup to measure geometry errors of the shaft thrust surfaces and journals.

Before the squareness of the thrust surfaces can be measured, the journal bearings must be aligned the to the datum axis of the roundness tester. The rotational datum axis is provided by a 4R Blockhead air bearing spindle as shown in Figure 1. Alignment requires iterative adjustment between the lower journal via centering and the upper journal via tip-tilt. Using this approach, the eccentricity between the two journals must be less than $0.2\ \mu\text{m}$ and is often less than $0.1\ \mu\text{m}$. A displacement of $0.2\ \mu\text{m}$ on the upper indicator represents 0.7 arc seconds angular motion.

The journal indicators are left in place to monitor alignment during subsequent tests. Squareness of the thrust to the journals is measured by two sensors on the thrust plate. The four sensors measuring the shaft simultaneously provide roundness of the journals, circular flatness of the thrusts, parallelism of one thrust to the other, and squareness of each thrust to the journals.

Essential to achieving the sub-micrometer precision in these shafts is measurement and manufacture with ability to make fine adjustments of the alignment of the journals using a high resolution tip-tilt table.

DESIGN

A tip-tilt table suitable for angular alignment of the shaft on the 0.1 micrometer-level in the roundness tester and capable of adequate support in the grinding processes is shown in Figure 2. Two convex plates are bolted together and adjustment screws near the outside diameter are used to provide the angular motion.

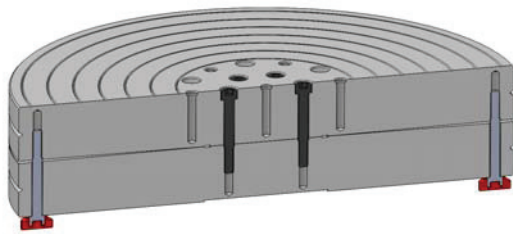


FIGURE 2. Cross-section of the tip-tilt table.

A grazing incidence interferometer (Tropel Flatmaster) was used to verify the form of the plates at the interface as shown in Figures 3 and Figure 4. The joint between the two plates is flat to 1 μm from the center to just beyond the inner bolt circle diameter. A flat joint at the center is critical to maintain joint stiffness. This convex form results in ± 10 arc seconds travel.

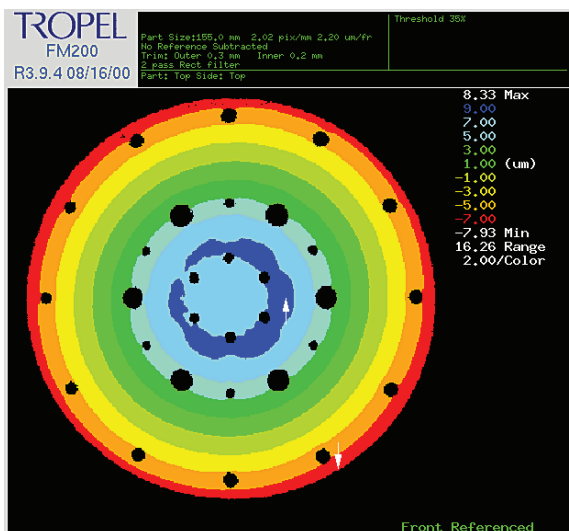


FIGURE 3. The convex top plate provides angular motion when an adjustment screw is tightened.

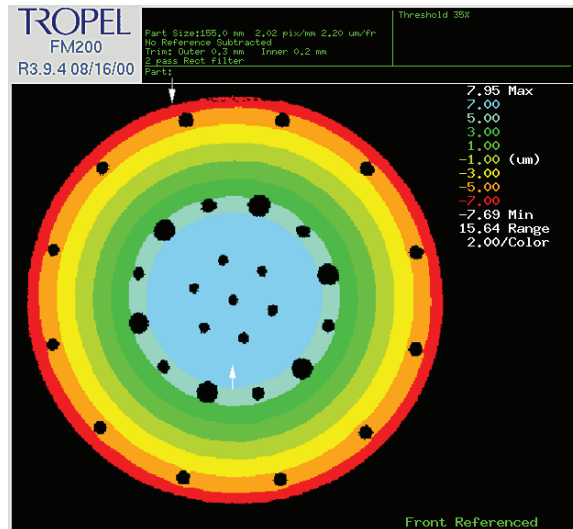


FIGURE 4. The convex bottom plate with flat surface at the joint.

RESOLUTION

A 180 mm long cylindrical square mounted to the table is measured at two axial locations to determine the pivot point of the table and realize the angular unit. With a separation between the two indicators of 178 mm, the upper sensor indicated 3.5 times more displacement than the lower one. As a result, the pivot point is located 22 mm below the joint between the two convex plates. If a 250 mm long shaft is placed on the table, each 60 nm of displacement at the end of the shaft is equal to 1 arc second of tilt.

The resolution of the tip-tilt table is demonstrated using a minimum incremental motion test. The test is performed using the smallest motion that can be repeated reliably three steps forward and three steps reversed. Actuation is performed manually using a single adjustment screw while monitoring the indicated displacement of the 180 mm cylindrical square.

The tip-tilt table demonstrates 0.2 arc second resolution as shown in Figure 5 with no lost motion or hysteresis.

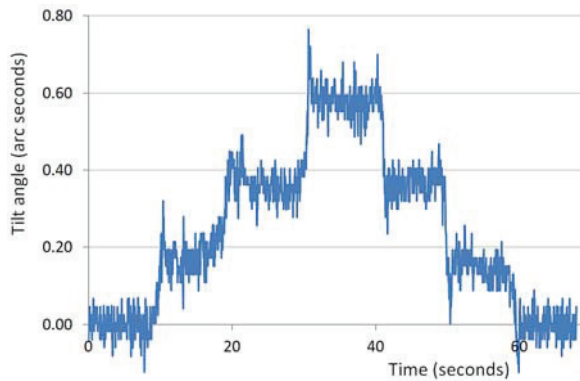


FIGURE 5. Minimum incremental motion test of the tip-tilt table is 0.2 arc seconds.

STATIC STIFFNESS

A finite element model is used to estimate the static stiffness of the tip-tilt table. A 1 Newton load is applied at the 145 mm bolt circle diameter of the adjustment screws. This is the approximate force of a single screw with 0.1 N•cm of tightening torque.

As shown in Figure 6, the screw force tips the table. If a 90 mm diameter part is centered on the table, angular motion of the part $0.05 \mu\text{rad}$. As a result, the static stiffness of the table is estimated at $2.9 \text{ N}\cdot\text{m}/\mu\text{rad}$.

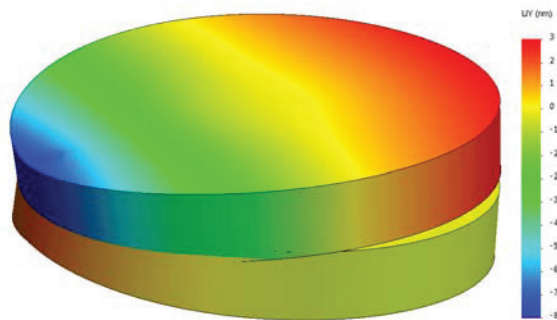


FIGURE 6. Finite element model of the deflection of table due to 1 N load.

DYNAMIC STIFFNESS

The finite element model is also used to estimate the dynamics of the table. The first two modes of vibration are a double mode represented by Figure 7 which occurs at 2,221 Hz. This is remarkably high for a metrology table and desirable for use as an adjustable machining table.

As is often the case, the first mode shape looks very similar to the deflection in Figure 6. In addition, the next two unique mode shapes are shown Figure 8 and Figure 9.

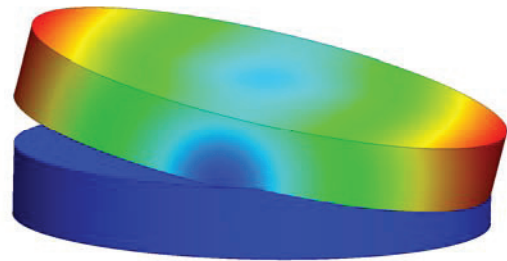


FIGURE 7. Finite element model of the first two modes at 2,221 Hz and 2,222 Hz.

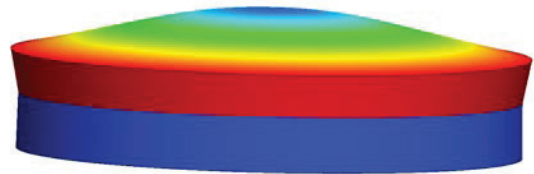


FIGURE 8. Third mode occurs at 3,615 Hz.

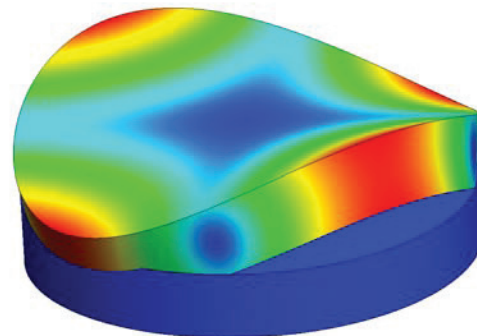


FIGURE 9. The next two modes occur at 4,848 Hz.

An experimental modal analysis was conducted using an accelerometer and roving impact hammer. Enough measurement locations were used so as to determine the various mode shapes. The magnitude of the driving point frequency response function is shown in Figure 10. The experimental results confirm the high first natural frequency and demonstrate outstanding damping. The high damping could be a benefit of the flatness of the surfaces at the bolted interface. The results are summarized and compared to the finite element results in Table 1. All of the predicted resonances are within 6% of the experimentally measured data.

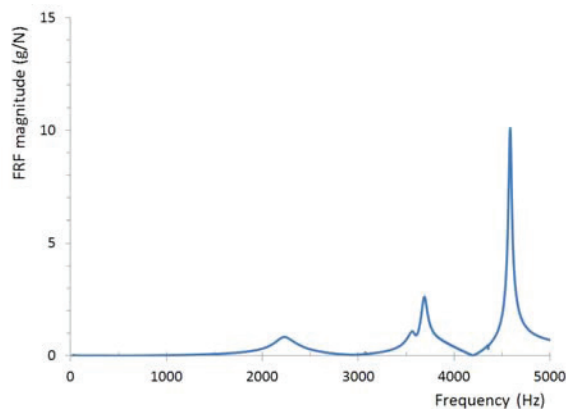


FIGURE 10. Modal analysis result shows the first mode at 2,210 Hz with outstanding damping.

TABLE 1. Comparison of the first six resonances predicted by finite elements and extracted from experimental data.

Mode	Frequency (Hz)		Difference	Description
	Exp	FEA		
1	2210	2221	1%	X tilt
2	2320	2222	-4%	Y tilt
3	3560	3615	2%	Defocus
4	3680	3615	-2%	Defocus
5	4580	4848	6%	Oblique astigmatism
6	4580	4848	6%	Vertical astigmatism

CONCLUSION

A tip-tilt table provides angular motion in two degrees of freedom with 0.2 arc second resolution. Many traditional designs suffer from stiction, hysteresis, lost motion, low stiffness, and low resolution. A flexure based design is demonstrated to overcome these issues. The high resolution is critical to measure the necessary squareness on precision air bearing rotors and stators. The exception static stiffness and dynamic response make the tip-tilt table also suitable for grinding adjustment tables.

TESTING THE EARLIEST SUB-MICROINCH SPINDLES

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Professional Instruments Co, Hopkins, MN

BACKGROUND

Proving sub-microinch axis of rotation error motion in 1962 required a sub-microinch test ball, unique indicators, and a robust test stand; combined with skills, patience and a deep understanding of the basic issues at hand.

In those days a “sub-tenth” (under 100 microinches) spindle was considered state of the art and a claim of less than one microinch was regarded with justifiable skepticism. But on the other hand there was a need for gyroscopes, optics, and other workpieces to be made to microinch-level tolerances and since axis error motions were a limiting factor there was motivation to devise a new mechanism for pure rotation. After exhausting the capabilities of ball bearing spindles and exploring oil-lubricated conical designs, a dual-hemisphere air bearing spindle was developed and now the challenge was to demonstrate extraordinary rotational accuracy in the days before low-noise cap gages and computerized centering.

MOTIVATION

The immediate application was as a workhead for ID grinding and a specific goal was uniform elastic symmetry of the rotor in order to avoid two-lobe radial error motion when carrying a heavy chuck and workpiece. Thus a test procedure was designed to show any irregularity in the bolted joints, the air films, or the rotor shaft itself.

HARDWARE

Figure 1 shows the arrangement for demonstrating radial and tilt accuracy under a balanced radial load. The spindle is secured to a heavy base and one-inch diameter test balls are flange mounted at both ends. Displacement indicators are robustly fixed to the base and arranged for measuring radial and tilt displacements.

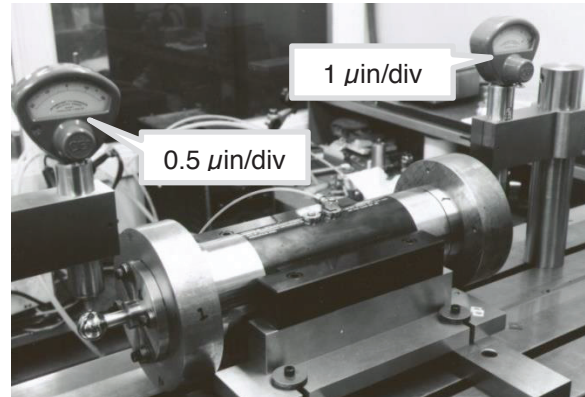


FIGURE 1. Demonstrating radial and tilt accuracy under load.

Figure 1 shows what we believe to be the world's first sub-microinch spindle. Because the Zeron was needed for machining applications, it had to maintain accuracy under load and this required an isoelastic rotor. For this test a ten-pound weight was attached to each end of the spindle rotor so that the steady force of gravity explored for any non-uniform flexing of the rotating shaft and its bolted joints.

In this case there was no drive influence, just coasting after hand-starting rotation at a few revolutions per minute.

INDICATORS

Displacements were measured with diamond-tipped Mikrokators; the one on the left is graduated in half-microinch increments and was interpolated to a fifth of a graduation and the one on the right has one-microinch graduations. These taut-band instruments, being frictionless, respond to nanometer-level displacements, but only if robustly mounted on heavy arms and posts. (The downside of high mechanical amplification is a high spring rate and this is why the structural loop must be stiff.) This old-school test rig inspired future designs to be built for stiffness; the watchword is that a good metrology setup should be capable of taking a cut by swapping the indicator for a lathe tool. This holds true even with non-contact gages since vibration of a probe holder can be confounded with axis error motion.

TEST BALLS

The test balls were lapped to less than a microinch out-of-roundness using procedures developed in-house for producing electrostatic gyroscope spherical rotors.

Centering the ball on the axis of rotation called for skill and patience: Final adjustments were made by preferential tightening of the four 10-32 socket head cap screws. Ball bearing washers under the screw heads reduced the granularity of the adjustments.

And, once the ball is centered, measurement must be prompt before thermal drift moves it off again. Contending with thermal drift is tedious but it vividly drives home the distinction between TIR and error motion.

INTERPRETING THE TEST RESULTS

It takes a while for a new observer to relate to the way runout drifts while the axis of rotation holds steady, and so to be convincing, the test is repeated by re-centering the ball so that the combined error of the spindle, out-of-roundness of the ball, and eccentricity results in under one microinch total indicator reading.

It's also a culture shock to realize that a mechanical indicator – of all things – is capable of seeing nanometer-level displacements.

Because of his work in lapping gyroscope spheres, Harold Arneson [Figure 2] was comfortable in the microinch world, but visitors, especially those skilled in the art of precision manufacturing, needed a little time to come to grips with direct observation of a millionth of an inch.



FIGURE 2. Harold Arneson demonstrates simultaneous measurement of axial, radial, and tilt error motions of a Zeron dual-hemisphere air bearing spindle.

AXIAL ERROR MOTION

Axial accuracy is important for machining optics, and in this unique arrangement axial motion is measured at both ends of the spindle (Figure 2).

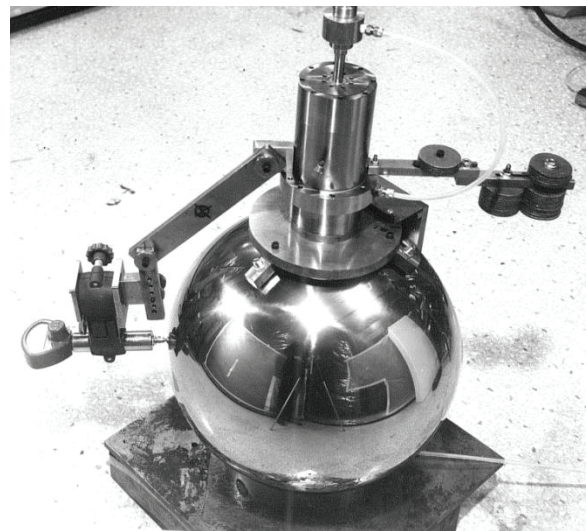


FIGURE 3. The roundness of a 17 inch ball is being measured relative to the axis of rotation of a Zeron spindle. (Shown here without insulation on the overarm.)

In a different arrangement (Figure 3), the out-of-roundness of a large sphere was measured by adjusting the spindle axis to pass through the center of the ball. (This ball became the rotor of an air bearing gimbal for a test platform for the Apollo moon landing program.) This job required tilt error motion at the sub-tenth-microradian level.

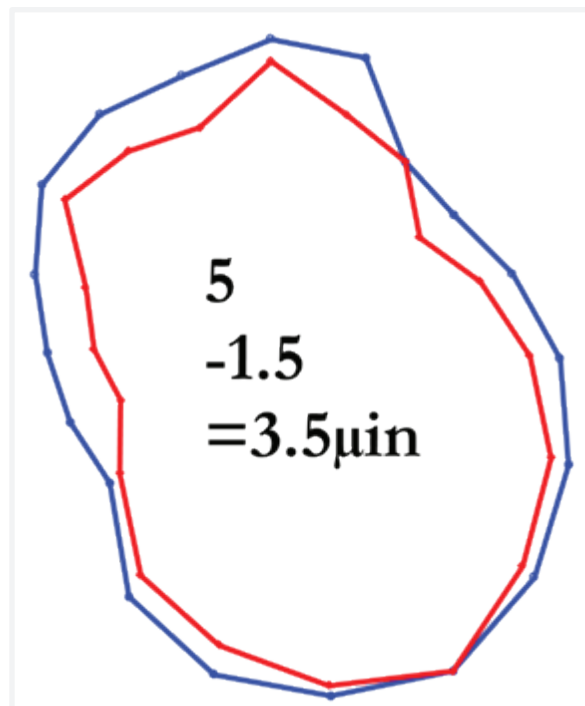


FIGURE 4. Spindle error motion was separated from out-of-roundness of the ball by indexing the spindle stator 90° and noting that the error stayed with the ball. The receiving inspector offered that it was “probably OK”.

The original data points for Figure 4 were hand-plotted on K & E polar coordinate graph paper from values carefully observed while hand-rotating the indicator. The present figure is the Excel presentation of those values.

The spindle error was sorted out from the out of roundness of the ball using the standard procedure. This process is aided by the fact that the error motion of an air bearing spindle is generally in the form of a smooth, two lobed egg shape; and the same is more or less true of the balls.

FIGURE 5. Description of error separation per Harold Arneson [1]

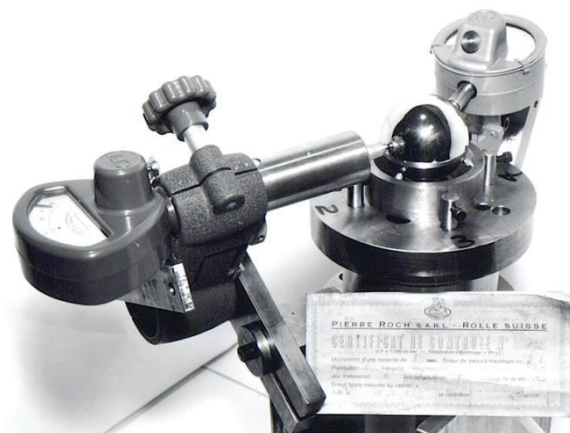


FIGURE 6. The arrangement for calibrating a number of 2-inch diameter, sub-microinch test balls.

Figures 3 and 5 show indicator support structures which are limp compared to those seen on Figures 1 & 2 and this is OK only because the indicator in this case is a special “Feathertouch” Mikrokator having an elaborate additional mechanism for constant spring rate as well as tip pressure adjustable to as low as one gram.



FIGURE 7. Close-up view of a half-microinch mechanical indicator.



FIGURE 8. Zeron ad from the 1960's.

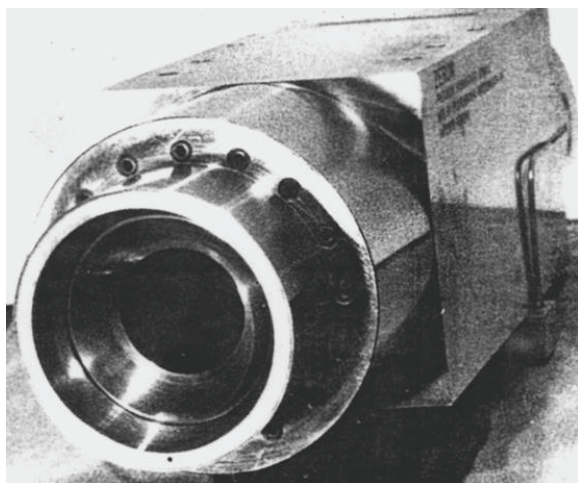


FIGURE 9. A Zeron stator showing the front hemispherical, step-compensated air bearing and its mounting arrangement.

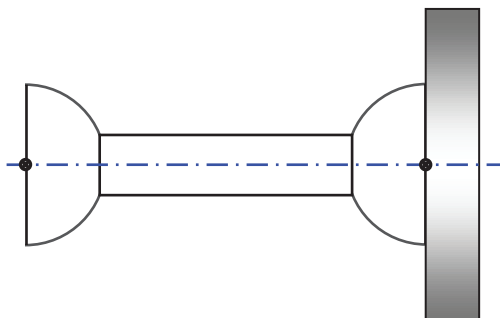


FIGURE 10. An advantage of a dual-hemisphere arrangement is that the bearings are self-aligning.



FIGURE 11. A disadvantage is that for overhung loads the rotor shaft needs to be long and this compromises radial stiffness.

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ITEM	QUANTITY	DESCRIPTION	
1	1	Zeron spindle in accordance with vendor's quotation dated 6/14/62, signed by A. A. Onnesorge and supplementary telegram dated 6/21/62, from Harold Arneson, "Specifications for Air Bearing Spindle" dated 6/19/62, and sketches #1 and #2 by T. G. Lewis are attached to and made a part of this order by reference.	7,500 ea.
Confirming phone order to Mr. Harold Arneson by E. T. P. Kennedy on 6/29/62. TILLED MAR 18 1963 Jra 7496 Hood			
MARK AND SHIP TO: E. I. DU PONT DE NEMOURS & COMPANY ORDER NO. ML-14361 DESTINATION: Mechanical Development Lab. 101 Beech Street Wilmington, Delaware Best Way SHIP VIA Wilmington, Delaware E. T. P. Kennedy PLEASE ENTER OUR ORDER AS SPECIFIED ABOVE. SUBJECT TO CONDITIONS AND INSTRUCTIONS LISTED ON BOTH THE FACE AND REVERSE SIDE OF THIS PURCHASE ORDER. E. I. DU PONT DE NEMOURS & COMPANY			

FIGURE 12. When the word got out about a spindle with extraordinary accuracy, Du Pont ordered one which made its way to Oak Ridge where it evolved into the famous "Y-12 Spindle".

CONCLUSIONS

The early days of spindle testing are worth examining because the basic principles are out in the open and the issues and approaches offer useful guidance for modern designs.

REFERENCES

[1] Private papers of Harold Arneson

OBLIQUE INCIDENCE STITCHING INTERFEROMETRY FOR MANUFACTURING HIGH PRECISION FLATS

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INTRODUCTION

The need for final form metrology under gravity loading (in the use orientation) is often encountered in the manufacture of large optical reference surfaces and artifacts. This is primarily due to the difficulty in predicting the self-weight deflection at the nm level due to the uncertainty associated with representing the actual boundary conditions in a finite element model. The primary metrology technique for the inspection of such artifacts is Fizeau interferometry, wherein the part under test (PUT) is compared to an optical reference of known shape (often referred to as the transmission flat or TF). In a typical Fizeau setup, the optical axis is arranged to be horizontal, i.e., the PUT is oriented vertically. However, in many applications, the use configuration of the PUT is horizontal, i.e., perpendicular to the gravity vector and the final form is desired in this orientation. This is achieved by either orienting the entire interferometer vertically or introducing a fold mirror (Figure 1).

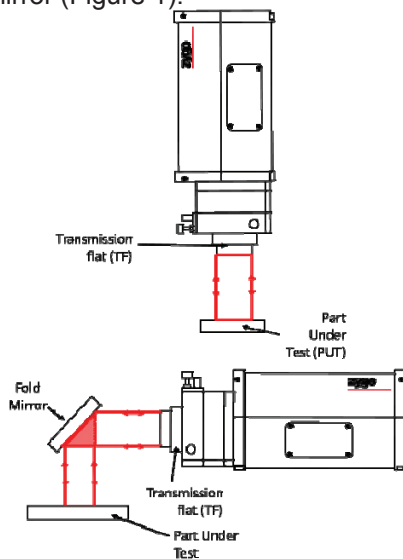


FIGURE 1 - Vertical test orientation direct and folded

For small apertures this is relatively straight forward but for larger apertures, >150mm, this can become a challenge due to the gravity

induced deformation on the shape of the transmission flat (the reference for the measurement) or the fold mirror.

This paper describes an interferometric measurement method that addresses the issues described above. This method is based on a standard 150mm aperture Fizeau interferometer for the inspection of parts significantly larger than the aperture of the interferometer. This is achieved by tilting the Fizeau interferometer at an angle to create an elliptical beam with a major axis that can extend to >500mm (Figure 2). This is similar to the Wilson double pass oblique test for measuring low reflective surfaces[1].

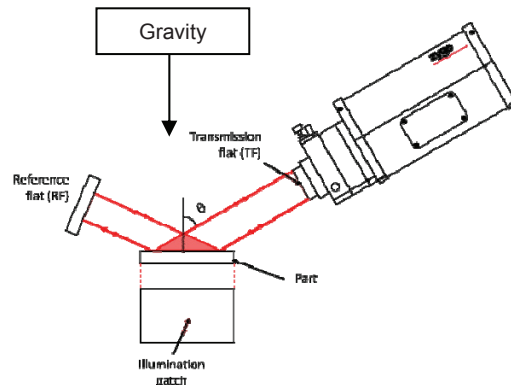


FIGURE 2 – Tilted test orientation

This paper will describe the measurement technique along with an uncertainty analysis that shows that the technique is capable of measuring surfaces of size ~0.5m square with an expanded uncertainty (k=2) of $\leq 10\text{nm}$ [2]. Measurement results of a 500mm square optical reference surface will be presented along with a description of the calibration procedure and associated mounts. This will include the measurement of the system wavefront error, calibration of the straight edge, error propagation analysis in the stitched direction and measurement uncertainty.

MEASUREMENT PROCESS

The PUT is presented to the interferometer in its horizontal use orientation. The entire optical surface of the PUT is measured by translating the optic in the minor axis direction while taking multiple overlapping sub-aperture measurements (Figure 3).

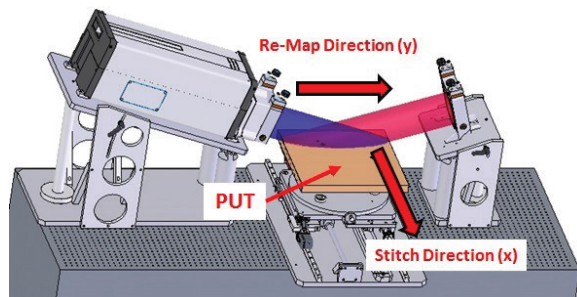


FIGURE 3 – System layout

This method takes advantage of the relatively small self-weight effects on the smaller TF and RF to interrogate a much larger optical surface. The overlapping measurements are then stitched together creating a surface height array that requires re-mapping in y and amplitude scaling. In the x direction the original pixel spacing of the interferometer is maintained. The surface height array requires re-mapping to correct for the foreshortening in the y direction. To minimize the interpolation errors the sampling of the new grid is based on the projected pixel spacing in the y direction. In general the surfaces measured are smooth. The surface height error from the interpolation routine will have a small contribution to the total measurement uncertainty. Amplitude scaling in z is a function of the angle of incidence and is described by equation (1) [3].

$$Z(\theta) = \frac{1}{4 \cos \theta} \quad (1)$$

If the dimensions of the PUT permit it can be advantageous to select the angle of incidence to be 60° as this will yield a scale factor equivalent to a normal incidence test.

TF AND RF CALIBRATION

TF/RF figure error and residual effects due to the tilt of the RF and TF are removed by calibrating them against another well calibrated artifact in place of the PUT. This is performed

using an optical flat or straight edge (SE) mounted on a deterministic support.

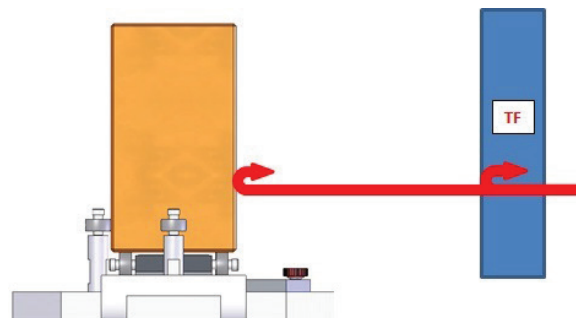


FIGURE 4 – SE horizontal Fizeau measurement

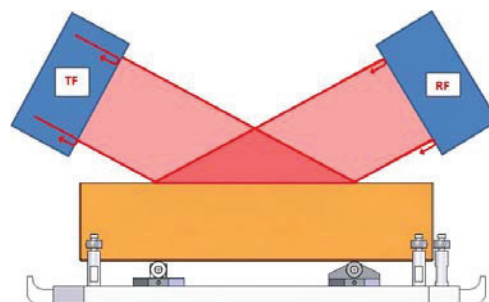


FIGURE 5 – Vertical Fizeau Measurement

The shape of the SE surface is initially established by a horizontal Fizeau measurement, i.e., with its surface vertical (Figure 4). The SE is then rotated to orient the surface perpendicular to the gravity vector, using the same deterministic support system (Figure 5). The SE is supported by 4 rollers (Tyre); two pivot and two are fixed. The supports are mounted on bearings to de-couple friction. The SE is placed within the optical cavity of the tilted interferometer creating a Fizeau cavity of the TF, SE and RF as shown in Figure 2. The resulting cavity produces a measurement $W(x,y)$ that includes the self-weight deflections of the $SE(x,y)$, $TF(x,y)$ and $RF(x,y)$.

$$W(x,y) = TF(x,y) + RF(x,-y) + SE(x,y) \quad (2)$$

The system wavefront error (SWE) from the TF/RF is determined by removing the SE measurement and the gravity sag contribution.

$$SWE(x,y) = W(x,y) - A \times SE(x,y') \quad (3)$$

A is the amplitude scaling defined by $A = 2 \cos \theta$ and $y' = y \cos \theta$ for describing the spatial compression in the direction perpendicular to the stitch.

SE self-weight deflection

The SE used has nominal dimensions of 443mm x 180mm x 99.25mm made from a low CTE material with a Modulus of Elasticity of 90.3GPa and a mass density of 2530Kg/m³. The calibrated surface is 443mm x 180mm. The support condition and artifact geometry is designed such that its self-weight deflection can be accurately determined by FEA modeling and/or analytical methods. A similar technique was used by Estler [4] for measuring straightness errors of precision machines. Modeling a simply supported beam with a uniform load will yield the following to calculate the self-weight deflection.

$$EI \frac{d^2 y}{dx^2} = M = R1 \langle x - a \rangle - \frac{wx^2}{2} + R2 \langle x - b \rangle \quad (4)$$

$$y(x) = \frac{1}{24EI} \left[R1 \langle x - a \rangle + R2 \langle x - b \rangle + wx^4 \right] \quad (5)$$

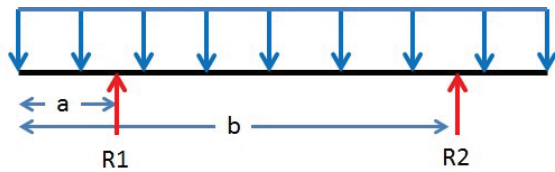


FIGURE 6 - Free body diagram of simply supported beam

It was determined during the design of the SE to limit the amount of self-weight deflection. This was accomplished by optimizing the stiffness (aspect ratio 5:1) and positioning the support points to balance the deflection at the edge and center (Figure 7 where L is the beam length). This is the support used for calibrating the TF/RF. This shape was removed from the SE system error file.

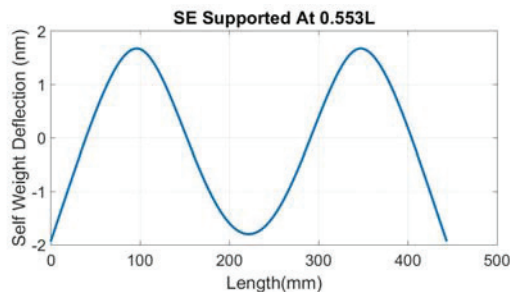


FIGURE 7 – Analytical calculation for the SE self-weight deflection with the optical axis vertical

To verify that these calculations are representative of the actual gravity deformation three measurements were taken at different support distances (0.408L, 0.553L and 0.697L) with the optical axis vertical (Figure 8). These measurements were performed using the oblique test described in figure 2 at an angle of incidence of 72.5°. Figures 8A through 8C show single slices through the full area dataset for three different support locations.

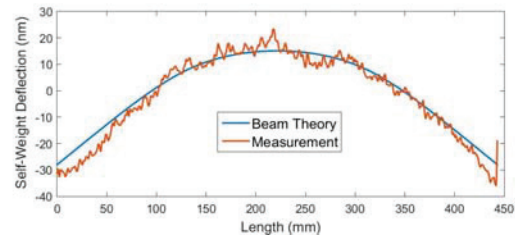


FIGURE 8A – Self-Weight Deflection at 0.408L

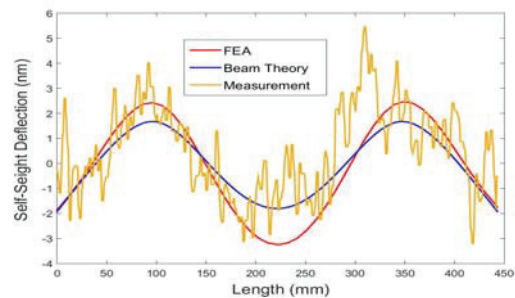


FIGURE 8B – Self-Weight Deflection at 0.553L. Configuration used for TF/RF calibration

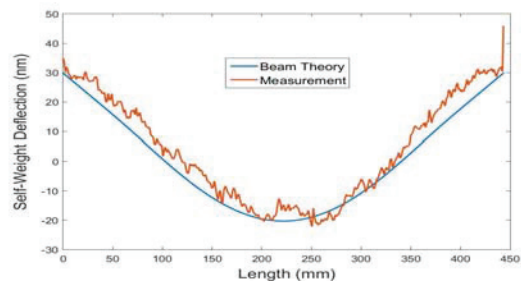


FIGURE 8C – Self-Weight Deflection at 0.697L.

The difference between the measured deflections and theory show reasonable agreement. Figure 8B includes the FEA as this is what is used for the SE self-weight deflection. Excluding the mid-spatial frequency errors of the SE the correlation is within 2 to 3 nm. The differences between FEA and beam theory are

attributed to the exclusion of shear deformation from the analytical formula.

SE self-weight deflection – un-deformed shape

Calibration of the SE surface, i.e. measurement of the un-deformed shape is performed in the horizontal orientation (Figure 4). The SE is supported on one of its sides with the surface under test oriented vertically, i.e. with the optical axis horizontal. The surface is measured in a conventional large aperture Fizeau interferometer. Measurement and analysis shows that the localized deformation induced by the support points has minimal impact on the measurement of the un-deformed shape. Figure 9 shows a finite element map resulting in approximately 11nm of support induced deformation. The FEA can be verified by measuring the SE in two orientations. This is done by taking data and then flipping the part 180° and re-measuring. Subtracting the two measurements will decouple the figure and mounting effects (Figure 10).

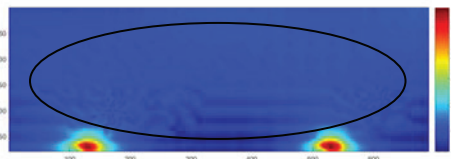


FIGURE 9 – FEA of the SE in horizontal configuration. PV=11.1nm

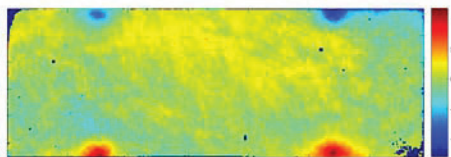


FIGURE 10A – Measurement difference of SE horizontal measurement

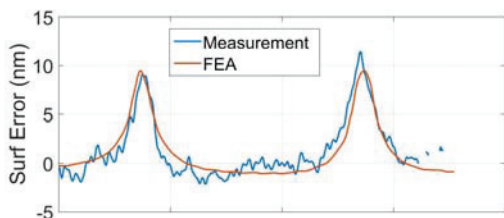


FIGURE 10B – Profile through the support point local deformation

While there are some differences between the FEA model and actual measurement the

correlation is acceptable. The contribution of the support deformation is also minimal as the bulk of the deformation lies outside the elliptical measurement region (Figure 9).

MEASUREMENT ERROR

Dominant uncertainty in the TF/RF calibration can be attributed to the environment (thermal stability) and uncertainty of the calibration of the SE. Another source of uncertainty is the contribution from stitching. Bray [5] discusses some of the error sources and how second order aberrations stitch. The linear stitch used here will propagate errors in one direction only. Small uncertainties (nm level) in the second order functions will amplify as the square of the aperture. It can be shown the rotationally varying errors can be driven to zero [6]. Evans [7] described errors relating to the test surface not being focused, the instrument transfer function and the Talbot effect. This has also been discussed by Bray [8] analyzing PSD (Power Spectral Density) effects for different cavity lengths. Another error source is knowledge of the angle of incidence (AOI). This will affect two aspects of the measurement; the amplitude scale and the spatial scale. The surface height error is proportional to the error in the measurement angle. For a 0.25° error in AOI the uncertainty in amplitude for a 100nm feature will be 2.3nm at a nominal measurement angle of 72.5°. For correcting surfaces using a sub-aperture polishing process the AOI knowledge is critical. Lateral scaling is also affected with the same 0.25° error for a 150mm interferometer contributing to an uncertainty in surface feature location of up to 7mm. This is a significant difference and will have adverse effects on the sup-aperture polishing process. To alleviate this effect fiducials are typically used within the measurement aperture to relate the alignment of the metrology coordinate system to the part.

MEASUREMENT OF A 500MM FLAT

The data presented here is for a 500mm square light-weighted flat. Three bi-pod flexures provide a stable three-point mount against a dynamic load application. The self-weight deflection at three points was large relative to the requirement at just under 1μm (1000nm). Correction for this deflection (and any residual errors) was applied through a sub-aperture polishing process. Data for which was obtained from the stitching system. 26 apertures were stitched at a nominal angle of incidence of 72.5° in order to cover the entire optical surface. The

measurement is performed 4 times indexing the part 90° each time to account for systematic errors. The part support stage and alignment controls are integrated within the measurement software MetroPro®. This allows automated data acquisition over a typical measurement duration of 8 hours for each position of the 500 x 500mm PUT. So the linear position along the stitching (x) direction for each aperture location is obtained from a linear encoder attached to the part stage and subsequently incorporated into the stitching routine. To ensure the part is aligned correctly with the TF/RF the tip/tilt stage is adjusted to auto null the fringe pattern at each of the 26 positions. Figure 11 shows the actual measurement set up for the 500mm optical flat.

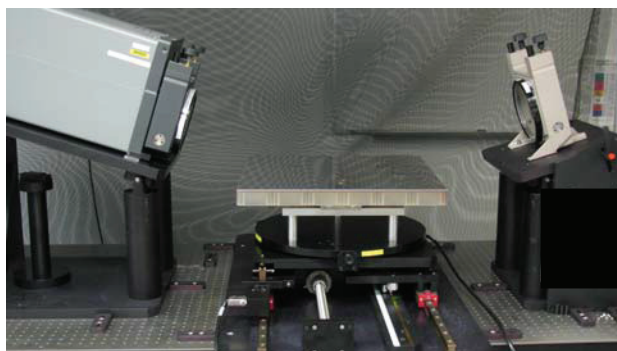


FIGURE 11 – 500mm optical flat measured at 72.5°

Measurement and uncertainty

The measurement process is designed to identify any systematic errors based on four measurements taken at 90° orientations. Any systematic errors will be evident in the stitched direction only. This is because the errors propagate in a single axis. Figures 12A through 12D show the measurement for each of the four rotations. They are all rotated back to a common orientation and plotted on the same scale.

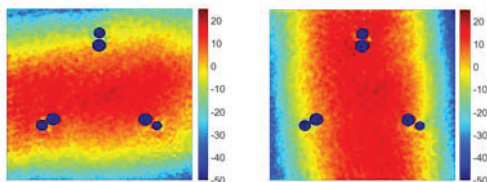


FIGURE 12A 0°

FIGURE 12B 90°

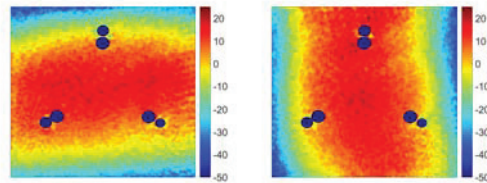


FIGURE 12C 180° FIGURE 12D 270°

It is obvious there is a systematic error in astigmatism. Averaging will drive these errors to zero (Figure 13).

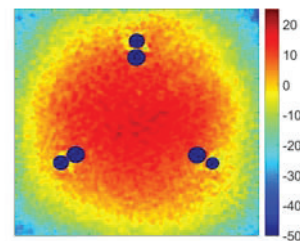


FIGURE 13 – Average of 4 rotations PV=60nm

Knowing the amount of astigmatism in each of the 4 rotations a correction file can be created and applied to the SWE map. A corrected SWE map can be used and the average re-calculated (Figure 14).

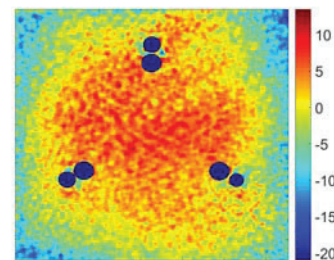


FIGURE – 14 Corrected measurement of 500mm optical flat PV=34.4nm and 4.75nm rms

The amplitude of the astigmatism corrected over the 150mm aperture was 4nm. The propagation of astigmatism couples with power. This is from the asymmetric projection of the oblique test. When the average map is re-computed the power is reduced by approximately 16nm.

The uncertainty matrix (Figure 15A) is a representation of the uncertainty at each measurement location (pixel) within the map. This map includes contributions from the standard deviation of the four rotations, the scale error and shearing of the data sets during

processing. The scale error contribution is based on an angle uncertainty of 0.1° . The shearing error stems from the residual misalignment between the data sets when the averaging is performed. This uncertainty was corrected for a small sample using the t distribution. For this analysis with 4 degrees of freedom the t value used was 5.841 for a one sided distribution at 99.5%. This value was based on the

$$mean + t \frac{\sigma}{\sqrt{n}}$$

(σ is the standard deviation and n

is the number of measurements). The 99.5 percentile was chosen as an extreme limit to identify what can be achieved with stitching metrology over an extended measurement period.

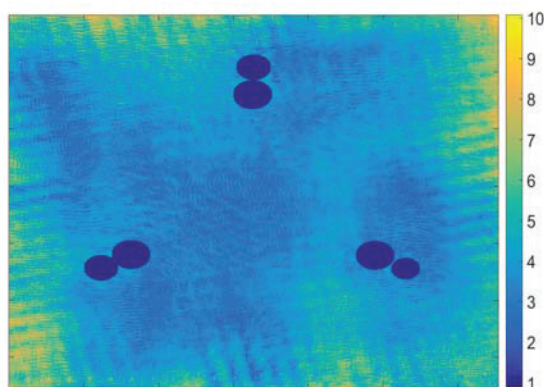


FIGURE 15A - Uncertainty Matrix (k=2)
Max = 10.07nm mean = 4.1nm 99.5th
percentile = 9.4nm

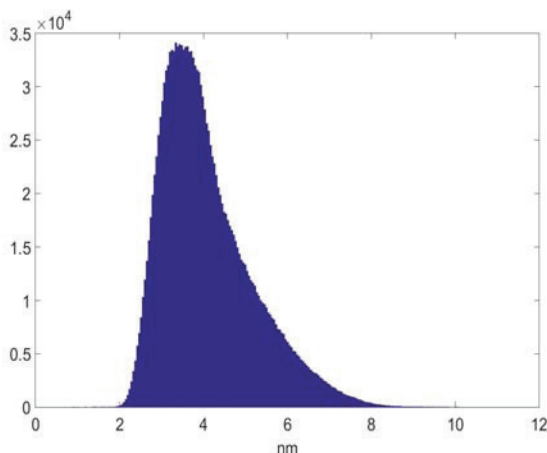


FIGURE 15B – Histogram of the uncertainty matrix Max = 10.07nm mean = 4.1nm 99.5th
percentile = 9.4nm

CONCLUSION

Using a smaller aperture interferometer at an oblique angle is a viable technique for stitching large optical flat surfaces. Measuring the system wavefront error, TF/RF error, with a known artifact can be achieved to nm level. Careful attention to the environment is critical as the measurement duration can be long for larger surfaces. Expanded measurement uncertainties of <10nm can be achieved.

ACKNOWLEDGMENT

The author would like to thank Chris Evans for the initial process verification and analysis [7], Vivek Badami and Jim Soobitsky for input on the hardware design and Aleks Estrin for metrology analysis support.

REFERENCES

- [1] Wilson, I.J. Double-pass oblique-incidence Interferometer for the inspection of non-optical surfaces. *Applied Optics*. 1983; 22: 1144-1148
- [2] Widdershoven, I., Donker, R. L., and Spann, H. A. M. Realization and calibration of the "Isara 4000" ultra-precision CMM. *Journal of Physics: Conference Series*. 2011; 311:012002
- [3] Zygo Application note – AN-0010 "Grazing Incidence Testing" 2008
- [4] Estler, W. Tyler Calibration and use of optical straightedges in the metrology of precision machines. *Optical Engineering* 1985 Vol. 24 No. 3 pg. 372-379
- [6] Evans, C., Kestner R. Test optics error removal. *Applied Optics* March 1996; Vol 35 No. 7 pg. 1015-1021
- [5] Bray, Michael *Stitching Interferometry and Absolute Surface Shape Metrology: Similarities SPIE Optical Manufacturing and Testing*, Vol.4451 2001
- [7] Evans, Chris Zygo R&D Tech Memo, August 2010
- [8] Bray, Michael *Stitching Interferometry: Side effects and PSD*, SPIE Optical Manufacturing and Testing III 1999, Vol. 3782 pg. 443-452

COMPUTATIONAL SPECTRAL IMAGING IN CHROMATIC CONFOCAL METROLOGY

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INTRODUCTION

Chromatic confocal imaging (1, 2) is a non-contact single-point distance measuring technique that benefits from all the advantages offered by confocal imaging such as eliminating the out-of-focus ambient light originated from points on the object other than the one being imaged. Additionally, in contrast to conventional confocal microscopy, chromatic confocal imaging eliminates the need for a vertical scanning mechanism by use of chromatic coding. In a chromatic confocal probe, the objective lens of a confocal microscope is replaced by a dispersive lens and a spectrometer is placed behind the confocal pinhole. The wavelength at which the intensity output of the spectrometer peaks can be related to the height of the object at that point. Although due to chromatic coding no vertical scanning is required, but chromatic confocal imaging is a single-point measuring technique and lateral scanning of the probe across the surface of the object is still necessary.

A new chromatic confocal metrology system is introduced in this paper that does not require any lateral scan. This system is based on a spectral imaging system capable of finding the wavelength information of whole field of view all at once. Therefore confocal imaging of a large surface area could be done at high speeds.

The spectral imager consists of a dispersive lens system which is intentionally designed to produce high amount of chromatic aberration. The image location of the dispersive lens system is highly wavelength dependent. For a fixed object point, conjugate image location depends on the wavelength content of the object. In conventional and chromatic confocal imaging techniques, the height of the object point is measured, respectively, based on the vertical location and the wavelength at which the

maximum intensity occurs at the pinhole. Our spectral imager works in an analogous manner but it maps the height of the object based on its conjugate image location or based on the location at which it forms a sharp image. Depending on the object height and the wavelength of illumination, a sharp or blurred image would form on the camera detector. The location of focused image and/or the amount of defocus for different illuminations can be used as a clue to calculate surface profile of the object. The calculation is done for whole field of view of the lens system simultaneously.

The design and implementation of the spectral imager, as well as the design of the new confocal metrology system, are discussed in this paper. The spectral imager consists of a dispersive lens system and an image processing tool. Images formed by the dispersive lens system are captured and analyzed by the image processing tool to find the spectral information of the object. The spectral information could be then used for finding the surface profile of the object which is the next step of this research.

CHROMATIC CONFOCAL METROLOGY SYSTEM

A schematic of the introduced chromatic confocal metrology system is depicted in figure 1. The object is illuminated by a source with known spectral characteristics and image(s) of the whole object is captured by a spectral imaging system. The spectral imaging system, which is the key component to the introduced metrology system, is designed in a way to be capable of locating the focused image for every point of the object. The location of focused image for every point is a function of its distance from the imaging system and therefore can be used as a clue to calculating the surface profile of the object. This process is done for whole

field of view at once so there's no need for lateral scanning. The concept and design of the spectral imager is described in following sections of this paper.

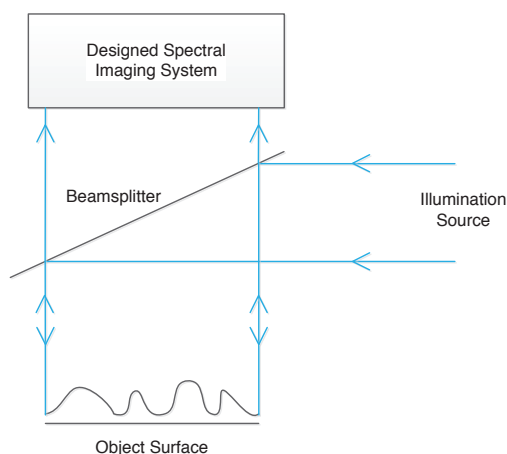


FIGURE 1. Schematic of the introduced chromatic confocal metrology system

DISPERSIVE IMAGING LENS SYSTEM DESIGN

As mentioned earlier, a main component of the spectral imager is the imaging lens system. The ideal lens system ought to fulfill two criteria: it should be highly dispersive and yet it should still form good-quality images. When designing an imaging system, designers usually try to avoid all types of aberrations including chromatic aberration. Here, we are taking advantage of the chromatic aberration to separate and disperse the spectral information of the object over its focal distance. Therefore, the chromatic aberration is desired and aimed to be maximized during the design. In fact, the larger is the chromatic aberration, the better wavelength resolution can be achieved by the spectral imager. However, except for chromatic aberration, other aberrations are undesirable and aimed to be minimized in order to maintain the quality of the images.

With above criteria, a triplet imaging lens system was designed. The specifications of the designed lens system are summarized in table 1 and a layout of the system can be seen in figure 2. The lens system produces a 3.96 mm separation between F (486.13 nm) and C (656.27 nm) image locations.

Due to its dispersive characteristics, dispersive lens system has a focal volume, Instead of a focal plane. As a result, the imaging system doesn't have a unique image location; each

TABLE 1. Specifications of the designed dispersive lens system (wavelength dependent values are reported at d wavelength)

Specification	Value
Magnification	-1.07
Field of view	8 x 8 mm ²
NA in object space	0.11
Axial RMS spot size	53 μ m
Effective focal length	40 mm
Object to image distance	151 mm
Distance between F and C image locations	3.96 mm

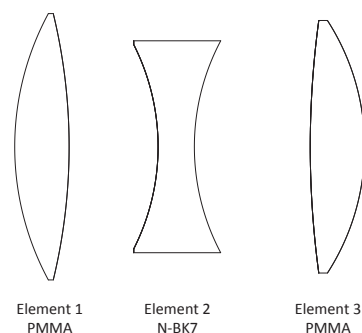


FIGURE 2. Layout of the designed dispersive lens system

wavelength has its own image location. Since the imaging system is completely calibrated, any location in front of the imaging system can be associated to a wavelength. Although the whole spectral information of the object would exist on every image taken at any location, only the spectral information corresponding to one wavelength would form a sharp and focused image on that plane. This specific wavelength is the wavelength associated to that location. For instance, the image location corresponding to blue color (F wavelength, 486.13 nm) is 67.3 mm away from the last lens for the designed system. If an image is acquired by the camera at this location, although all the existing colors in the object contribute to the formed image but it's only the blue image which is sharp and focused at this location. All the other colors form blurred and defocused images. We need to distinguish between the focused and defocused images at every location in order to find the spectral information of the object. It is, therefore, possible to correlate the sharpness of the image with wavelength information of the object and the problem of calculating the spectral information of the object comes down to finding the location(s) of focused image for every pixel of the object.

Each focused image corresponds to a wavelength content of that pixel

Now that the imaging system is designed we need an algorithm to analyze images formed by the lens system and calculate the spectral information of the object. Two categories of algorithms have been used here and will be discussed in next section.

IMAGE PROCESSING TOOL

The designed dispersive lens system in the previous section spreads the wavelength information of the object over a volume and its images can be used to compute the spectral images of the object. As mentioned before, if the focused image location(s) can be found for every pixel, the wavelength information can be computed consequently. Finding the location of sharp image is also the problem which must be solved in 3D imaging (3). In 3D imaging, the imaging system is ideally non-chromatic but the object is three dimensional. Each pixel of the object is blurred based on its distance from the imaging system. If the location of focused image is found for each pixel of the object, then the depth map of the object is determined and the problem is solved. Similar scenario occurs in our case where spectral information of a two-dimensional object is of interest and imaging is done by a dispersive lens system. Therefore, we are facing a similar issue as 3D imaging. The difference is that here it is the wavelength information of each pixel that makes its image blurred not its depth. However, the concept is the same and some of the algorithms that are used in 3D imaging can be applied to our spectral imaging system.

Two categories of algorithms are of interest here: shape from focus and shape from defocus. These two well-known algorithms are widely used for depth recovery and 3D reconstruction (4-6). As discussed earlier, with some modifications these algorithms can be used here to analyze the images produced by imaging lens system. The basic idea behind shape from focus algorithm is to acquire several images of the object at different distances from the imaging system. Then a criterion function is defined to measure the sharpness of each acquired image. For any pixel on the object, the critical points of the criterion function determine the locations of focused images and each location corresponds to a specific wavelength. The basic idea behind shape from defocus, on the other hand, is to obtain just a few images of the object. Instead of trying to locate the focused images, the

algorithm determines the amount of defocus or blur for these few images and calculates the wavelength information based on that. The images obtained from the designed imaging system are analyzed using these two algorithms to find the wavelength information of the object.

Shape From Focus (SFF)

Shape from focus (4, 5, 7) is a passive depth estimation technique that first searches for the focused image and then uses the location of focused image along with the known camera parameters to calculate the depth. The main challenge here is locating the focused image and it's done by taking a sequence of images at different locations and measuring their sharpness. If this algorithm is used, a vertical scanning mechanism is required to capture a sequence of images at different distances from the lens system but once the images are captured, the spectral information of whole object can be calculated. So no lateral scanning is needed.

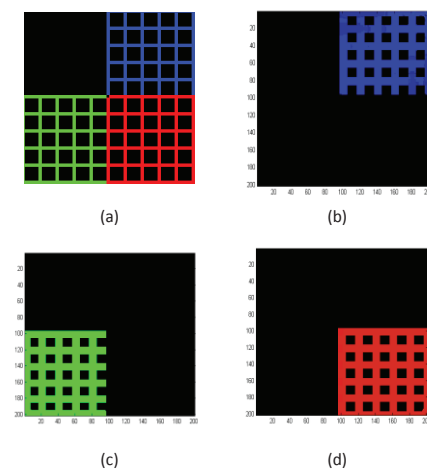


FIGURE 3. (a) Object contains information at 486 nm (shown as blue), 587 nm (shown as green) and 656 nm (shown as red) wavelengths. Spectral images of the object in (a) are shown at (b) 486 nm, (c) 587 nm and (d) 656 nm calculated by SFF algorithm.

SFF algorithm was applied to images obtained from lens system shown in figure 2. Figure 3a shows the object and its wavelength content. It contains information at three different wavelengths: F (486 nm, shown as blue), d (587 nm, shown as green) and C (656 nm, shown as red). 100 images were taken at 50 μm steps. Modified gray-level variance (MGLV) (8) was

used as the criterion function for focus measure. The maximum focus measure determines the location of focus. Depending on the wavelength content, focus measure function corresponding to each pixel peaks at a specific location. The values of focus measure function of pixel number (79,142) using 100 images are plotted in figure 4 as an example. This particular pixel which is on the green part of the object and has d wavelength content, is in focus at image number 59 as can be seen from figure 4. As a matter of fact, focus measure function corresponding to every pixel which contains information at d wavelength would show a peak at this location. So we can find the spectral image of the object at d wavelength by finding every pixel which is in focus at the location of image number 59 or equivalently every pixel which its focus measure function peaks at this location. Spectral images calculated by SFF are shown in figure 3c-d.

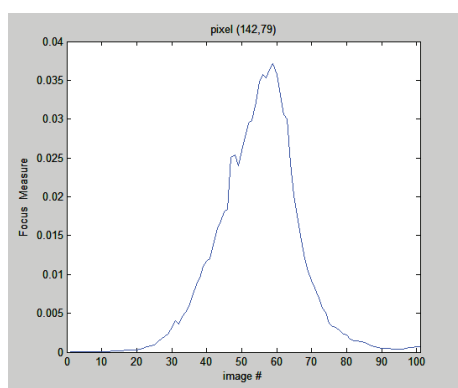


FIGURE 4. Focus measure function of pixel number (79,142) using 100 images

Shape From Defocus (SFD)

Shape from defocus (3, 4, 6, 9) is another passive depth estimation technique. It takes two or more images as its input. Instead of looking for the focused image as with SFF, SFD determines how much these images are defocused. Depth information can be calculated consequently. The mathematical modeling presented in (9) was used in this paper. Two images were used as inputs to the algorithm. The resulting spectral images are also shown in figure 5. Since just two images are used for the spectral computations, the need for vertical scanning is alleviated when using SFD compared to SFF but less accurate results have been obtained.

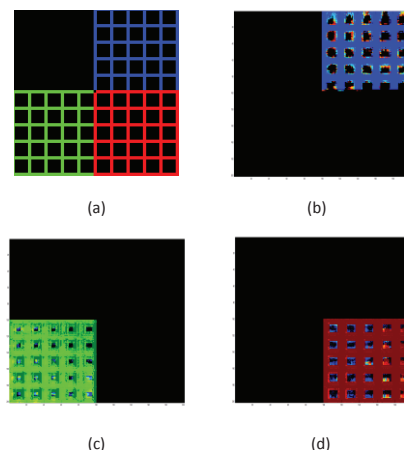


FIGURE 5. (a) Object contains information at 486 nm (shown as blue), 587 nm (shown as green) and 656 nm (shown as red) wavelengths. Spectral images of the object in (a) are shown at (b) 486 nm, (c) 587 nm and (d) 656 nm calculated by SFD algorithm.

CONCLUSION

A computational spectral imaging technique was introduced to be used for chromatic confocal metrology. The design and functionality of a spectral imager was discussed in this paper. The introduced spectral imager is capable of finding the location of sharp image for every point across the field of view. The location at which the imaging system forms a sharp image of the object point is correlated to the distance of that point from the imager and can be used to calculate the surface profile of the object. The introduced technique computes the spectral information of whole field of view all at once and therefore offers high rates for surface metrology of large areas.

REFERENCES

1. Blateyron F. Chromatic confocal microscopy. Optical Measurement of Surface Topography: Springer; 2011. p. 71-106.
2. Garzón J, Gharbi T, Meneses J. Real time determination of the optical thickness and topography of tissues by chromatic confocal microscopy. Journal of Optics A: Pure and Applied Optics. 2008;10(10):104028.
3. Favaro P, Soatto S. 3-d shape estimation and image restoration: Exploiting defocus and motion-blur: Springer Science & Business Media; 2007.
4. Xiong Y, Shafer SA, editors. Depth from focusing and defocusing. Computer Vision and Pattern Recognition, 1993 Proceedings

CVPR'93, 1993 IEEE Computer Society Conference on; 1993: IEEE.

5. Ens J, Lawrence P. An investigation of methods for determining depth from focus. Pattern Analysis and Machine Intelligence, IEEE Transactions on. 1993;15(2):97-108.

6. Ziou D, Deschênes F. Depth from defocus estimation in spatial domain. Computer Vision and Image Understanding. 2001;81(2):143-65.

7. Nayar SK, Nakagawa Y. Shape from focus. Pattern Analysis and Machine Intelligence, IEEE Transactions on. 1994;16(8):824-31.

8. Pertuz S, Puig D, Angel Garcia M. Analysis of focus measure operators for shape-from-focus. Pattern Recognition. 2012.

9. Favaro P, Soatto S. A geometric approach to shape from defocus. Pattern Analysis and Machine Intelligence, IEEE Transactions on. 2005;27(3):406-17.

SURFACE PROFILE MEASUREMENT FROM AN OBLIQUE DIRECTION USING A CHANGE IN REFLECTION ANGLE

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INTRODUCTION

Measurement sensitivity vs. range is a trade-off relationship. Therefore, a stitching method is used to accurately measure the surface profile of a wide area. With the stitching method, measured surface profiles with high sensitivity at narrow areas are synthesized by software processing. However, as the number of measured narrow areas is increased, measuring time becomes longer. Therefore, while measuring many narrow areas from above the surface, the surface under test is exposed to dangers, such as falling objects, contact with measuring devices, and so on, over a long period of time. Since creating a large-scale product is very expensive, damage during the course of measurement is not allowed. Therefore, in this paper, we propose a method to measure a planar surface with a wide region

from an oblique direction using a change in reflection angle. With the proposed method, nothing is placed above the surface under test, and there is therefore no risk of damaging the surface under test. The proposed method determines the angular error of the surface under test with high sensitivity by the optical lever method. The angular error is integrated to calculate the shape error. In this study, the conditions of the measurement object are assumed as follows: Shape error with a depth of a few microns and a diameter in the order of ten cm, is distributed over several tens of meters.

PRINCIPLE

Figure 1 shows the proposed optical arrangement. Initially, the emitted laser beam is incident on the surface under test. The beam reflected by the surface under test reaches the

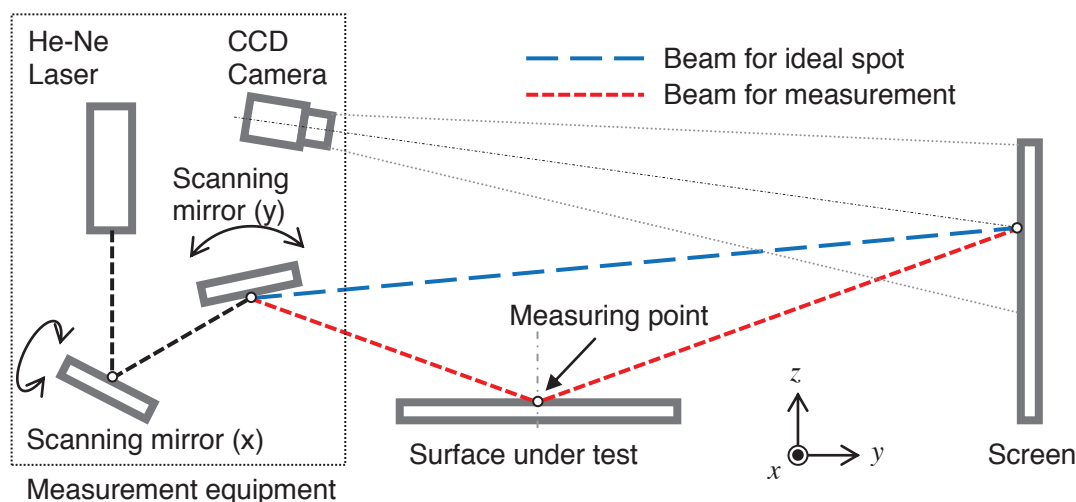


FIGURE 1. Proposed optical setup.

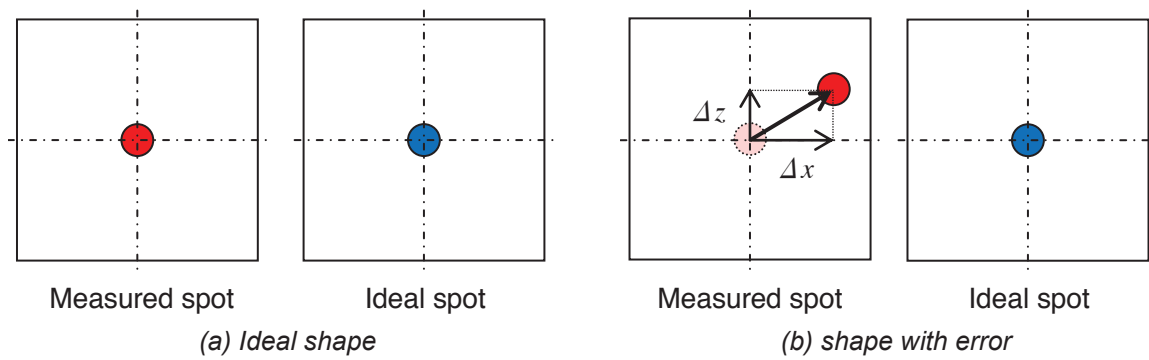


FIGURE 2. Measured spot images.

screen (red dashed line shown in fig.1). The beam on the screen includes information on the surface under test. If the surface under test has a shape error, the position of the beam on the screen shifts from the ideal position. The ideal position of the beam on the screen can be calculated from geometrical information of the measuring equipment and the surface under test. However, if the geometric relationship between the measurement equipment and the surface under test is not correct, measurement results are distorted significantly.[1] Therefore, we propose to radiate the beam (blue long dashed line shown in fig.1) to the ideal reflection position on the screen. With this method, if the screen is set at a position only slightly different from the correct position, it is possible to cancel the effect of the placement error.

Measurement equipment

The measurement equipment is composed of a laser scanning unit and an image acquisition unit. The scanning unit moves measuring points and ideal spots. In the acquisition unit, the CCD camera is mounted on a pan-tilt head that is controllable by a computer. Spot images are acquired by the CCD camera. Figure 2 shows the measured spot images.

Process to obtain spot images

For one measurement point, a set of ideal spot and the measurement spot are obtained. To obtain the ideal spot, the laser beam is directly radiated toward the ideal position on the screen by the scanning unit. The direction of the CCD camera is controlled by the pan-tilt head to move the position of the ideal spot to the center of the image. The measured spot is then obtained by fixing the pan-tilt head in the direction. If the ideal shape is measured, the position of the ideal spot agrees with the measured spot as

shown in fig. 2(a). As a shape with error is measured, the position of the measured spot shifts from the position of the ideal spot according to the shape error as shown in fig. 2(b). Δx and Δz are value related to the inclination error on the x and the y direction at the measurement point, respectively. The inclination error from the ideal shape of the measurement surface is calculated from the amount of shift of the spot. Therefore, the shape error is obtained by integrating the inclination error.

Relationship between the measurement error and the placement error of the screen

When the ideal shape is measured by the proposed method, the ideal spot agrees with the measured spot. However, the spots are for example shifted by a geometrical error of the screen position. Figure 3 shows a case of screen position with error that includes only parallel movement along the y direction. In the figure, the upper and the lower parts represent the top and the front views, respectively, of the proposed method. In the top view, the ideal spot agrees with the measurement spot along the x direction. The spot shift of the x direction represents the inclination error along the x direction of the surface under test. Therefore, the screen position is independent of the measurement error. Similarly, in the front view, the ideal spot and the measurement spot shift along the z direction. The amount of the shift of the spots is $\overline{DE}$. $\overline{DE}$ is expressed as follows.

$$\begin{aligned}\overline{DE} &= \overline{DI} - (\overline{EF} + \overline{FI}) \\ &= \overline{DI} - (\overline{HI} \tan \phi + \overline{BH} \tan \theta) \\ &= 2 \frac{\overline{HI}}{\overline{GH}} \overline{AG} \\ \therefore \tan \theta &= \frac{\overline{DI}}{\overline{BI}}, \tan \phi = \frac{\overline{EF}}{\overline{HI}}\end{aligned}$$



300 mm. The measuring line passes through point B, and is parallel to the x axis. Figure 4 shows the results of the simulation. The horizontal axis is the amount of shift of the screen, and the vertical axis represents the PV

$$\varepsilon = \tan^{-1} \frac{\overline{\text{DE}}}{\overline{\text{BI}}}$$

Position error of screen mm (X)	Proposed method (Y)	Conventional method (Y)
-300	0	-100
-200	0	-65
-100	0	-35
0	0	0
100	0	30
200	0	60
300	0	90

396

value of the error. The marks $\circ$ and $\bullet$ indicate the results of the conventional method and the proposed method, respectively. By the proposed method, the measurement error is nothing.

EXPERIMENTS AND RESULTS

Figure 5 shows the experimental setup. The light source is an He-Ne laser. The laser beam is scanned by galvano mirrors. To increase the spatial resolution of the measurement on the surface under test, the measurement equipment includes a zoom lens. The spot size is the smallest on the surface under test. Therefore, since the spot size on the screen is increased, image processing of the spot becomes resistant to noise. In the experiments, the distance from the measurement equipment to the surface under test is 1,000 mm, and the distance from the surface under test to the screen is 2,900 mm. Figure 6 shows the relationship between the proposed method and the conventional method, which is a non-contact three-dimensional measuring machine. Comparing the two results, the measurement error is less than $\pm 0.4\mu\text{m}$.

CONCLUSIONS

In this paper, a new method from an oblique direction is proposed to measure a planar surface with a wide region, and the validity of the proposed method is confirmed by experiments.

REFERENCES

- [1] Takashi Nomura, Kazuhide Kamiya, Seiichi Okuda, Keishin Yamazaki, Hatsuzo Tashiro, Kazuo Yoshikawa, Shape error measurement using ray-tracing and fringe scanning methods, Journal of the International Societies for Precision Engineering and Nanotechnology, 26 (2002) 30-36.

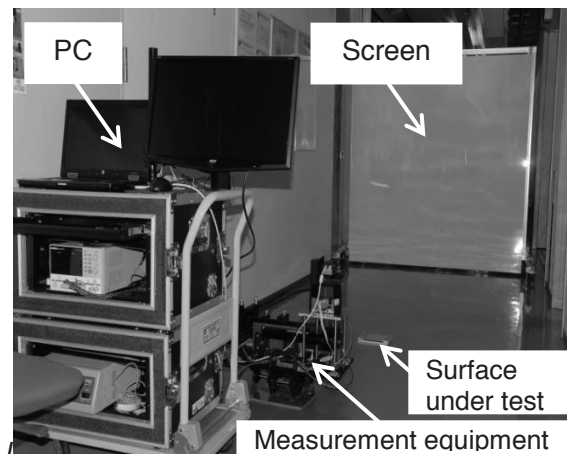


FIGURE 5. Experimental setup

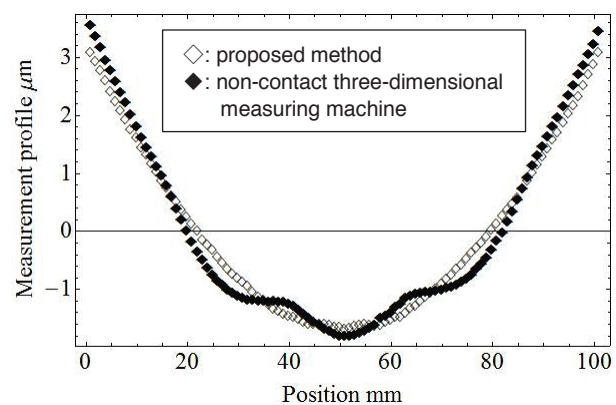


FIGURE 6. Comparison of the measurement results by the proposed method and other methods

NOVEL DISPLACEMENT SENSING TECHNIQUE UTILIZING KNIFE-EDGE DIFFRACTION

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INSTRUCTIONS

High precision stages have been extensively used in many machining and manufacturing facilities such as semiconductor processing, disk drive read/write testing, machine tool metrology and assembly line testing. Precision stages can be fabricated to have high positioning accuracy with a large dynamic range and bandwidth [1-6]. Displacement sensing technology plays the primary role for fast and robust positioning control and high resolution measurement relied on by high-tech manufacturing and metrology operations [6-11]. Although strain gauges and LVDT's are used in some stage applications, we restrict our focus to non-contact sensors. Non-contact sensors such as capacitive displacement sensors and optical sensors (laser interferometer, laser encoder and position sensitive detector (PSD)) are traditionally used in high precision stage applications due to their ability to perform dynamic motion characterization and fast and high resolution measurement [2,7] as summarized in Figure 1. Capacitive type sensors are the mostly widely used because they can detect motion at sub-nanometer levels directly and provide accuracy, linearity, resolution, and stability. Capacitive displacement sensor and PSD are nominally preferred over the laser interferometer and laser encoder because they are more compact and easily embedded into a precision stage. However, the capacitive sensors are limited to conductive targets and the attractive force between the target and the sensor probe should be considered for precision applications [12]. In addition, capacitive displacement sensors are relatively expensive. On the other hand, PSDs are commonly used because they are

inexpensive and capable of measuring lateral displacement in one or two dimensions. However, many issues about the position nonlinearity and sensitivity have been raised due to the beam size, the gap size between the elements, incident light intensity uniformity and aberration, a window in-place stray reflected light the optical alignment and the doping uniformity of each active area [2,13]. Those problems can be solved by implementing a look up table (LUT), but their limitation is their noise [11].

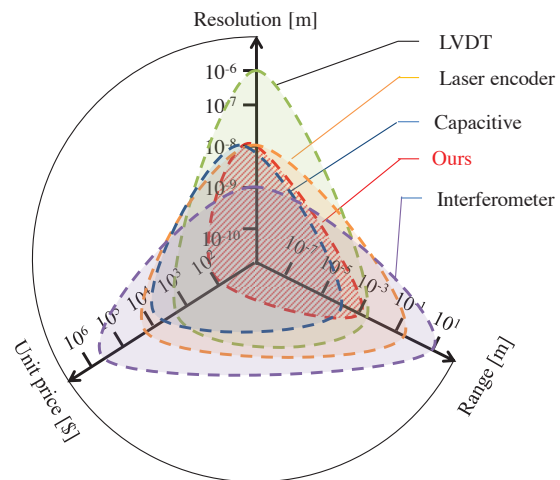


Figure 1. Comparison of various displacement sensors.

Moreover, the PSD suffers from sensitivity issues. It is well-known that the smaller the beam spot size is the higher the sensitivity. Therefore, the beam spot should be made as small as possible to improve the sensor sensitivity. However, the minimum beam spot size is limited because of the gaps that exist

between each photodiode, which are not sensitive to the beam spot [2] and are not always uniform.

The last few decades have seen significant effort to develop nanometer resolution displacement sensing instrumentation for high precision stages. Companies such as Physik Instrumente, Lion Precision and InSituTec, have been using the capacitive sensors integrated with flexure stages, and achieved sub-nanometer resolution.

The literature also demonstrates a number of efforts of to accomplish high-precision sensing for nanopositioners with other novel sensing methods. Tang first introduced the lateral comb drive in an electrostatic resonator, and has since been used numerous times in both actuator and sensing configurations [14-17]. However, a simple, inexpensive, noncontact high-resolution sensor that can be easily embedded into a high precision stage has not been well-documented. In this work, a simple and economic high resolution knife edge-based displacement sensor for machine tool metrology is presented. The edge diffraction model suitable for the proposed measurement apparatus is mathematically derived, and the effect of the parameters associated with the edge diffraction investigated. In addition, the fundamental limits of the linearity and resolution of the sensor is discussed by estimating the effects of edge topography of the knife edge on the knife edge diffraction of an incident wave based on Kirchhoff approximation.

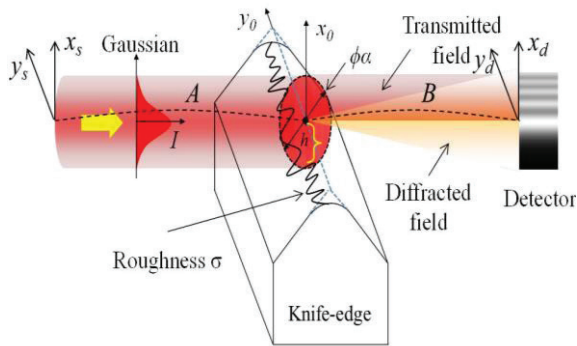


Figure 2. Knife-edge diffraction by randomly rough edge surface morphology.

THEORY: KNIFE EDGE DIFFRACTION

We recently introduced the knife-edge diffraction analysis method and created a deterministic model to characterize its diffraction phenomenon according to the edge roughness [18]. Assuming that a Gaussian beam with the beam diameter α is incident on the knife-edge, the total field to the

front of the knife-edge is simply the incident field and the beam propagates along the z axis as illustrated in Figure 2. The incident field $\vec{E}_s$ can be defined as [19],

$$\vec{E}_s(x_s, y_s, z_s) = E_s e^{-jk_0 A} e^{-\frac{x_s^2 + y_s^2}{\alpha^2}} \quad (1)$$

where, E_s is the amplitude of the incident field, k_0 is a wave vector in air, α is the beam width, and A and B are the distance between optics components. The first and second terms are an expression of the incident wave with Gaussian intensity distribution and the last term is for an aperture of the field. The incident field $\vec{E}_s$ creates constructive and destructive interference due to phase function with respect to the knife-edge displacement. The superimposed wave can be derived by using Fourier transform (FT) and the total field $\vec{E}_d$ (sum of transmitted and diffracted field) at the detector, can be obtained by applying inverse FT of the superimposed wave [18]. The total field to be measured along the z axis can be defined from the inverse FT relation of the incident field as,

$$\vec{E}_d(x_d, y_d, z_d) = \iint_{k\text{-space}} \frac{1}{(2\pi)^2} \times \iint_{\text{aperture}} \vec{E}_s \cdot e^{j\vec{k} \cdot \vec{r}_{i \rightarrow 0}} e^{-j\vec{k} \cdot \vec{r}_{0 \rightarrow d}} dy dx dk \quad (2)$$

where $\vec{r}_{i \rightarrow j}$ is a vector from i to j coordinate system.

Optical metrology systems can be extremely precise but are fundamentally limited by the surface quality of optics components. As an approximated model to a rough knife-edge surface is presented, phase difference between the transmitted and diffracted fields is related to the roughness, assuming that the roughness is randomly distributed in nature. The roughness on the knife edge boundary is assumed to comprise a nonzero mean with Gaussian probability density function (*pdf*) [18]. Based on this assumption, knife edge surface roughness is modeled in a *pdf* form as,

$$\vec{E}_d^R(x_d, y_d, z_d) = \iint_{k\text{-space}} \frac{1}{(2\pi)^2} \iint_{\text{aperture}} \vec{E}_s \cdot e^{j\vec{k} \cdot \vec{r}_{i \rightarrow 0}} e^{-j\vec{k} \cdot \vec{r}_{0 \rightarrow d}} \times pdf(h) dy dx dh dk \quad (3)$$

$$pdf(h) = \frac{1}{\sigma\sqrt{2\pi}} e^{-\frac{h^2}{2\sigma^2}}$$

where R represents a rough surface and h is the random height along the y_0 axis and σ is roughness, standard deviation of h as illustrated in Figure 2. Here, the roughness along the x_0 axis was only considered and independent of the y_0 axis.

$$I_d(y_0) = \langle \tilde{E}_d^R \rangle \times \langle \tilde{E}_d^R \rangle^* \quad (4)$$

Where, the total power I_d induced by the knife-edge at detector will be calculated by multiplying the total field and the conjugated total field at each detector plane. The asterisk (*) denotes a complex conjugate.

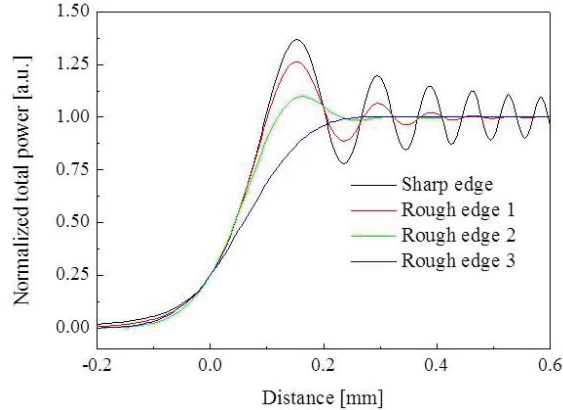


Figure 3. Total power curves according to edge surface roughness.

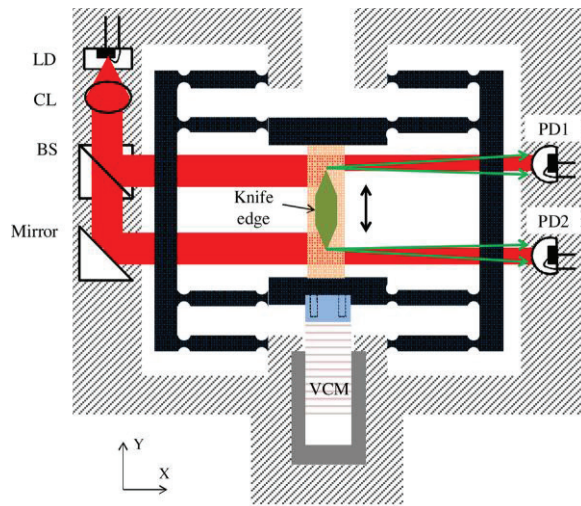


Figure 4. Schematic illustration of the proposed nanopositioning system.

The more the edge morphology is rough and dull, the more incoherent the diffracted field becomes because the phase between the transmitted and diffracted fields changes according to edge morphology conditions. The interference of the transmitted and edge-diffracted field can theoretically explain the general oscillatory behavior. Note that as the edge roughness increases, the amplitude oscillations decrease due to the destruction of the phase interference between the transmitted and diffracted fields as seen in Figure 3.

EXPERIMENT: DISPLACEMENT SENSOR

The proposed nanopositioning system consists of three parts: flexure mechanism, actuator and sensor as illustrated in Figure 4. The flexure mechanism was designed in a double compound notch-type with hinges. Each spring was designed with circular flexure hinges at both ends. The VCM was assembled with the flexure stage and aligned along the moving axis of the stage to mitigate Abbe error. The optical knife edge sensor (OKES) includes the laser diode (LD), collimation lens (CL), beam splitter (BS), mirror, double-side sharp knife edge and two lensed photodiodes (PD). The BS-to-knife edge and knife edge-to-PD distances were equally set to 50 mm and the distance between two parallel beams incident on the knife edge was set to 22 mm (knife edge width). The OKES utilizes the light interference in order to get high sensitivity. The LD light source is collimated with the CL and separated by a 50:50 at the BS. The reflected light is incident on an upper knife edge facet mounted on the stage which is oriented parallel to the traveling direction, and the transmitted light at BS is reflected off the mirror and incident on a lower knife edge facet. The two separated lights are partially transverse and diffracted at each knife edge and the interference signal is measured at each PD. The lensed PDs were used to achieve a longer working range and eliminate the effect on positioning errors induced by non-parallelism between the OKES and DCNFM. This photovoltaic signals at each PDs are then amplified using a differential amplification signal processing technique. Consequently, the displacement along the traveling direction can be detected simultaneously by Eq. (5),

$$V_{out}(y_0) = K \times \frac{V_{PD1} - V_{PD2}}{V_{PD1} + V_{PD2}} \quad (5)$$

where K is a sensitivity of the sensor, and V_{PD1} and V_{PD2} are out-puts measured at PD1 and PD2, respectively. It uses a difference of V_{PD1} and V_{PD2} divided by a sum of V_{PD1} and V_{PD2} , which can effectively eliminate the dependency of light intensity. This measurement method is considered to achieve the high resolution and sensitivity of the sensor because the interference is highly sensitive to the displacement of the knife edge.

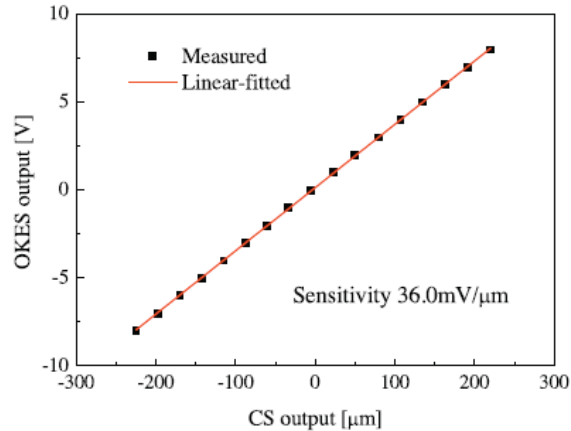


Figure 5. Calibration result: CS represents a capacitive type sensor.

In use of the lensed PDs, the measuring range became longer. The CS (10 nm resolution with 500μm range, Lion Precision) was used to calibrate the OKES and compare with its performance and control effectiveness in the OKES-based nanopositioning system. The analog circuit was created for PID controller, motor driver and sensor amplifier. The OKES measuring range was set similar to the CS working range to compare two sensors at the same condition. As seen in Figure 5, the OKES was calibrated by using the CS and showed a sensitivity of 36.0mV/μm with high linearity.

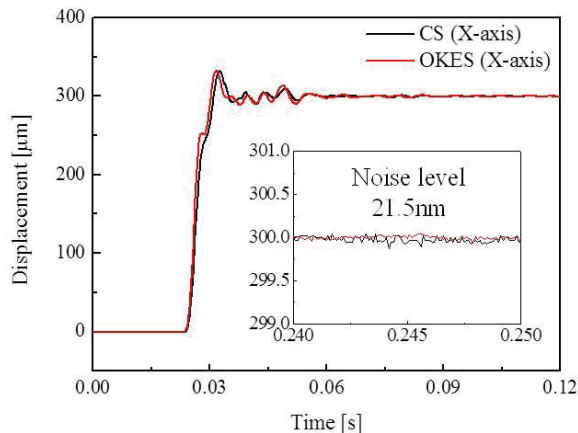


Figure 6. Step response curve.

A 300μm step test of the XY stage was performed as shown in Figure 6. The OKES and CS results showed good agreement and the steady-state positioning root-mean-square (rms) noise, which is also representative of the resolution, was 21.5nm along X.

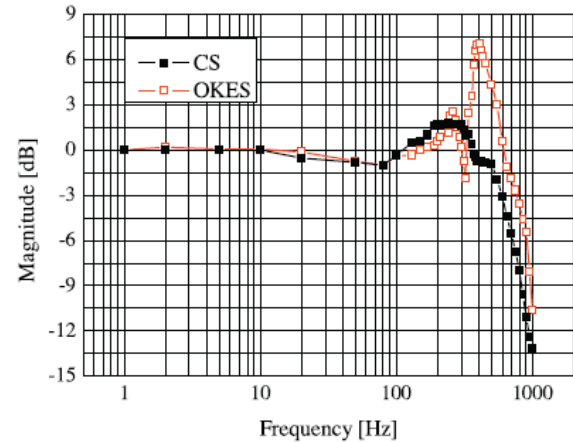


Figure 7. Calibration result: CS represents a capacitive type sensor.

The dynamic characterization of the stage was investigated by obtaining the frequency response curves as seen in Figure 7. Two sensors showed similar results below 100 Hz and a resonance peak of the stage at 280 Hz. However, the OKES showed a large second resonance peak at 400 Hz. On the other hand, the CS showed no significant peak around 400 Hz. It was thought that the first resonance peak is due to the flexural mode of the stage and second resonance peak is related to torsional mode of the stage because the flexure stage typically has a coupled motion with flexural and torsional modes. It was explained that the CS measures the capacitance change of an effective area (Ø3.2 mm) at a single point and it becomes less-sensitive to the torsional motion if it is installed along the driving axis with no offset distance. On the other hand, the OKES measures two points (22 mm interval) around the center of the stage and it can be sensitive to the torsional motion. As such, angular motion, especially for high frequency signals, has an influence on the OKES readout. This does not mean that OKES does not perform well. The OKES can measure the angular motions with high sensitivity that CS cannot measure.

CONCLUSIONS

In this research, we created a design principle of the OKES by deriving mathematical model and characterized the OKES performance in terms of working range, positioning accuracy, resolution, linearity, bandwidth, and control effectiveness with the nanopositioning system. The results showed that the OKES is capable of large range and nanometric resolution for single-axis or multi-axis operation, compact size and low cost,

and this sensor can be good alternative to the CS and optical encoders or even better sensor selection in nanopositioning applications.

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REFERENCES

- [1] Sumeet S Aphale, Santosh Devasia and S O Reza Moheimani, "High-bandwidth control of a piezoelectric nanopositioning stage in the presence of plant uncertainties," *Nanotechnology*, Vo. 19, 125503, 2008.
- [2] W. Gao, *Precision Nanometrology: Sensors and Measuring Systems for Nanomanufacturing*, Springer, 2010.
- [3] Smith, S., *Flexures: Elements of Elastic Mechanisms*, Taylor & Francis, London, England, 2002.
- [4] Michael A Cullinan, Robert M Panas and Martin L Culpepper, "Multi-axial MEMS sensor with integrated carbon nanotube-based piezoresistors for nanonewton level force metrology," *Nanotechnology*, Vol. 23, 325501, 2012.
- [5] Ki Woon Chae, Wook-Bae Kim and Young Hun Jeong, "A transparent polymeric flexure-hinge nanopositioner, actuated by a piezoelectric stack actuator," *Nanotechnology*, Vol. 22, 334401, 2011.
- [6] ChaBum Lee, Gyu-Ha Kim and Sun-Kyu Lee, "Design and construction of a single unit multi-function optical encoder for a six-degree-of-freedom motion error measurement in an ultraprecision linear stage," *Meas. Sci. Technol.* Vol. 22, 105901, 2011.
- [7] ChaBum Lee and Sun-Kyu Lee, "Multi-degree-of-freedom motion error measurement in an ultraprecision machine using laser encoder – Review," *J. Mech. Sci. Technol.*, Vol. 27, No. 1, pp. 141-152, 2013.
- [8] ChaBum Lee, Gyu-Ha Kim and Sun-Kyu Lee, "Uncertainty investigation of grating interferometry in six degree-of-freedom motion error measurement," *Int. J. Precis. Eng. Manuf.*, Vol. 13, No. 9, pp. 1509-1515, 2012.
- [9] Yusuke Saito, Yoshikazu Arai, Wei Gao, "Investigation of optical sensor for small tilt angle detection of a precision linear stage," *Meas. Sci. Technol.*, Vol. 21, 054006, 2010.
- [10] Akihide Kimura, Wei Gao and Satoshi Kiyono, "Design and construction of a surface encoder with dual sine-grid," *Int. J. Precis. Eng. Manuf.*, Vol. 8, pp. 20-25, 2007.
- [11] Gillmer SR, Smith RCG, Woody SC, Ellis JD, Compact fiber-coupled three degree-of-freedom displacement interferometry for nanopositioning stage calibration, *Meas. Sci. Technol.* Vol. 25, No. 7, 075205, 2004.
- [12] Fukuo Suganuma, Atsushi Shimamoto, and Kohichi Tanaka, "Development of a differential optical-fiber displacement sensor," *Appl. Opt.*, Vol. 38, No. 7, pp. 1103-1109, 1999.
- [13] H X Song, X D Wang, L Q Ma, M Z Cai and T Z Cao, "Design and performance analysis of laser displacement sensor based on position sensitive detector (PSD)," *Journal of Physics: Conference Series*, Vol. 48, pp. 217-222, 2006.
- [14] William C. Tang, Tu-Cuong H. Nguyen and Roger T. Howe, "Laterally driven polysilicon resonant microstructures," *Sensors & Actuators*, Vol. 20, pp. 35-32, 1989.
- [15] Yuen Kuan Young, Andrew J. Fleming, S. O. Reza Moheimani, "A novel piezoelectric strain sensor for simultaneous damping and tracking control of a high-speed nanopositioner," *IEEE/ASME Trans. Mechatronics*, Vol. 18, No. 3, pp. 1113-1121, 2013.
- [16] Ali Bazaei, Mohammad Maroufi, Ali Mohammadi and S. O. R. Moheimani, "Displacement sensing with silicon flexures in MEMS nanopositioners," *J. Microelectromech. Syst.*, Vol. 23, No. 3, pp. 502-504, 2014.
- [17] Gaurav Parmar, Kira Barton, Shorya Awtar, "Large dynamic range nanopositioning using iterative learning control," *Precis. Eng.*, Vol. 38, pp. 48-56, 2014.
- [18] Eugene Hecht, *Optics* 4th Ed., Addison-Wesley Longman Inc., 1987.
- [19] Bradley A. Davis and Gary S. Brown, "Diffraction by a randomly rough knife edge," *IEEE Trans. Antennas Propag.*, Vol. 50, No. 12, 2002.

IMPROVEMENT OF A PERIODIC ERROR COMPENSATION ALGORITHM BASED ON THE CONTINUOUS WAVELET TRANSFORM

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INTRODUCTION

Heterodyne displacement measuring interferometry provides important metrology for applications requiring high resolution and accuracy, such as in the semiconductor manufacturing industry and for linear stage calibration. Heterodyne Michelson interferometers use a two-frequency laser source and separate the two optical frequencies into one fixed length and one variable length path via polarization. Ideally these two beams are linearly polarized and orthogonal so that only one frequency is directed toward each path. An interference signal is obtained by recombining the light from the two paths; this results in a measurement signal at the heterodyne (split) frequency of the laser source. This measurement signal is compared to the optical reference signal. Motion in the measurement arm causes a Doppler shift of the heterodyne frequency which is measured as a continuous phase shift that is proportional to displacement. In practice, due to misalignment of optical components, component imperfections, and elliptical polarization, undesirable frequency mixing occurs which yields periodic errors [1-3]. Typically, 1st, 2nd and even higher order periodic errors occur, which correspond to the number of periods per fringe displaced, as shown in Figure 1. A displacement fringe corresponds to the wavelength divided by the interferometer fold factor which is determined by the interferometer setup. Ultimately, this error can limit the accuracy to approximately the nanometer level.

Many studies have investigated the measurement and compensation of periodic error, including frequency domain [4] and time domain approaches [5, 6]. For frequency domain

approach, the periodic error can be measured by calculating the Fourier transform of the time domain data collected during constant velocity target displacement. However, this method is not well suited to non-constant velocity profiles and sub-fringe positioning ranges. An alternate digital algorithm which can be applied in real-time for constant or non-constant velocity motions is also available for measuring and compensating 1st and 2nd order periodic error. But this method leaves 3rd and higher order periodic errors as residual errors.

In this research, a real-time continuous wavelet transform (CWT) based algorithm, which was developed in previous work [7], is improved and used to compensate 1st, 2nd and higher order periodic errors (modeled as pure sine signals). Moreover this algorithm can compensate errors in both constant and non-constant velocity motions.

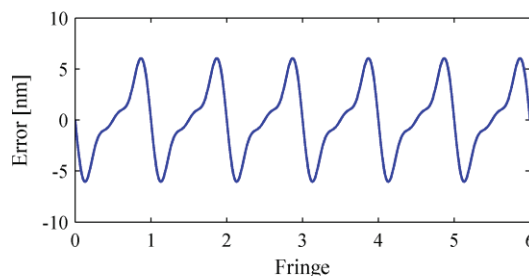


FIGURE 1. Example of 1st, 2nd, and 3rd order periodic error as a function of fringes. Typically, 1st order error has a larger magnitude than higher order errors.

CONTINUOUS WAVELET TRANSFORM

The wavelet transform can be used to analyze time series data that contains non-stationary (variable period) power at multiple frequencies [8]. Wavelet functions refer to either orthogonal or non-orthogonal wavelets. The choice of the appropriate wavelet transform (continuous or discrete) and wavelet function is based on whether the purpose of data analysis is detection or compression [9].

A wavelet function $\psi(t)$ is a finite energy function [10] with an average of zero,

$$\int_{-\infty}^{+\infty} \psi(t) dt = 0. \quad (1)$$

A wavelet family is generated by dilating the mother wavelet via the scale $s > 0$ and translating it via the location $u \in \mathcal{R}$. This series of wavelets can be expressed as

$$\psi_{u,s}(t) = \frac{1}{\sqrt{s}} \psi\left(\frac{t-u}{s}\right). \quad (2)$$

In this research, a continuous wavelet transform (CWT) is used to analyze the signal $x(t)$, with a wavelet function $\psi(t)$. For a one-dimensional signal $x(t)$, the CWT is defined as the convolution of $x(t)$ with a scaled and translated version of $\psi(t)$ via

$$\begin{aligned} Wx(u,s) &= \int_{-\infty}^{+\infty} x(t) \psi_{u,s}^*(t) dt \\ &= \int_{-\infty}^{+\infty} x(t) \frac{1}{\sqrt{s}} \psi^*\left(\frac{t-u}{s}\right) dt \end{aligned} \quad (3)$$

where $\psi^*(t)$ is the wavelet function, s is the scale, and u is the location. In the present work, the complex Morlet wavelet is used as the mother wavelet

$$\psi^*\left(\frac{t-u}{s}\right) = \pi^{-\frac{1}{4}} e^{i2\pi f_0 \frac{t-u}{s}} e^{-\frac{1}{2}\left(\frac{t-u}{s}\right)^2}. \quad (4)$$

In practice, Equation 3 must be converted from continuous to discrete:

$$Wx(n,s) = \sum_{n'=1}^M \left(x(n') \sqrt{s} \psi^*\left(\frac{(n'-n)\Delta t}{s}\right) \Delta t \right), \quad (5)$$

where $x(n)$ is the n^{th} discrete data point, ψ^* is the mother wavelet, M is the number of total data points in the signal, and Δt is the sampling time.

After applying the complex Morlet wavelet to the signal, the CWT result is a two-dimensional complex array. This array can be used to extract the CWT “ridge” and, therefore, the phase of the periodic errors. The ridge is the location where the CWT coefficient reaches its local maximum along the scale direction [11]; the coefficient is maximum when the analysis frequency equals the signal frequency [12]. The ridge and phase are

$$ridge(n) = \max \{ |Wx(n,s)| \} \quad \text{and} \quad (6)$$

$$\phi(n,s) = \arctan \left(\frac{\text{Im}(Wx(n,s_{ridge}))}{\text{Re}(Wx(n,s_{ridge}))} \right), \quad (7)$$

where s_{ridge} is the scale at the ridge, and Im and Re represent the imaginary and real parts of the CWT coefficient, respectively.

COMPENSATION ALGORITHM

The periodic error compensation, which can be processed in real-time, is depicted in Figure 2.

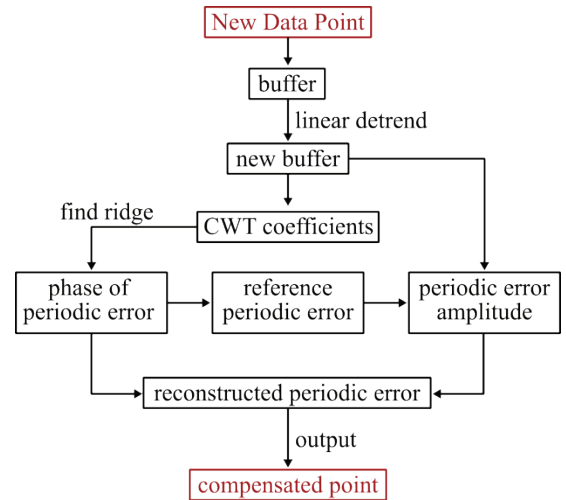


FIGURE 2. Calculations to implement the periodic error compensation algorithm.

Each time a new data point is obtained, an N -size array is populated with the last N data points. With this series of data, the CWT is computed at the last data point (i.e., location n in Equation 5 is fixed at the end of the array). Therefore, an array of coefficients at the end point along scales is obtained and the ridge can be determined at scale s_1 . This scale corresponds to the 1st order periodic error frequency. Because the scale is inversely

related to the frequency, the scale $s_i = s_1 / i$ corresponds to the i^{th} order periodic error frequency.

For each new data point, the ridge and phase is calculated, so the periodic error phase information is determined. Arrays for the reference j^{th} order periodic error are constructed,

$$r_j[1 \dots N] = \begin{cases} \sin(j\varphi(1)), \sin(j\varphi(2)), \dots, \\ \sin(j\varphi(N)) \end{cases} \quad (8)$$

We consider a general form of m order periodic errors,

$$\sum_{j=1}^m A_j r_j[1 \dots N], \quad (9)$$

where A_j is j^{th} order periodic error amplitude. Apply CWT linearity property to obtain equations:

$$\begin{cases} c_1 = A_1 d_{11} + A_2 d_{12} + \dots + A_m d_{1m} \\ c_2 = A_1 d_{21} + A_2 d_{22} + \dots + A_m d_{2m} \\ \vdots \\ c_m = A_1 d_{m1} + A_2 d_{m2} + \dots + A_m d_{mm} \end{cases}, \quad (10)$$

where c_i is the CWT result for the data array at scale s_i , and d_{ij} is the CWT result for reference j^{th} order periodic error at scale s_i .

The amplitudes can be solved and then the periodic error is reconstructed as

$$\sum_{i=1}^m A_i \sin(i\varphi(N)).$$

Finally, this result is

subtracted from the original data to determine the compensated displacement data point.

SIMULATIONS

Simulations were used to assess the validity of the CWT-based algorithm.

The algorithm was applied to a simulated constant velocity motion (50 mm/min) with 1st, 2nd, and 3rd order periodic error amplitudes of 4 nm, 2.5 nm, and 1 nm, respectively. The measured amplitudes are shown in Figure 3. The relative errors are 3.1%, 3.8%, and 8.9%, respectively. The compensated result is displayed in Figure 4. The root-mean-square (RMS) error is decreased by approximately 86.4%.

The algorithm was also verified using a simulated accelerating motion (30 mm/min²

acceleration, 4 nm, 2.5 nm, and 1 nm amplitudes for 1st, 2nd, and 3rd order periodic errors, respectively). The measured amplitudes are shown in Figure 5. The relative errors are 8.6%, 4.8%, and 10.8%, respectively. The compensated result is provided in Figure 6. The RMS error is reduced by approximately 80.1%.

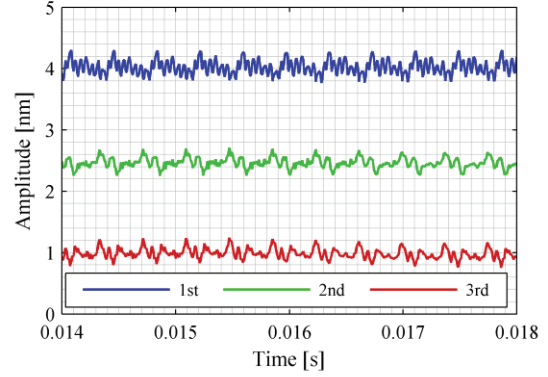


FIGURE 3. The measured amplitudes of 1st, 2nd, and 3rd order periodic errors in the constant velocity motion.

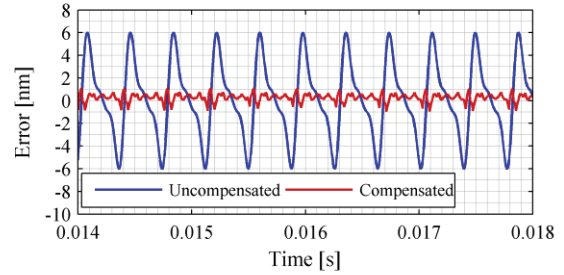


FIGURE 4. The result of periodic error compensation in the constant velocity motion.

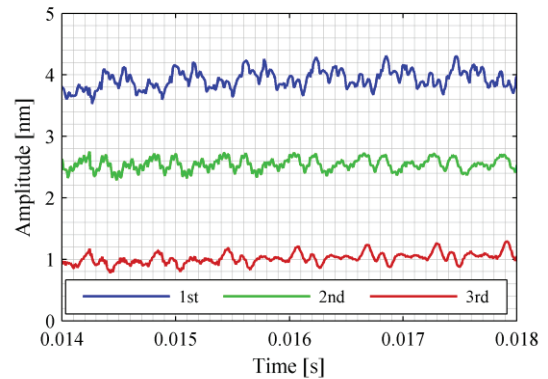


FIGURE 5. The measured amplitudes of 1st, 2nd, and 3rd order periodic errors in the non-constant velocity motion.

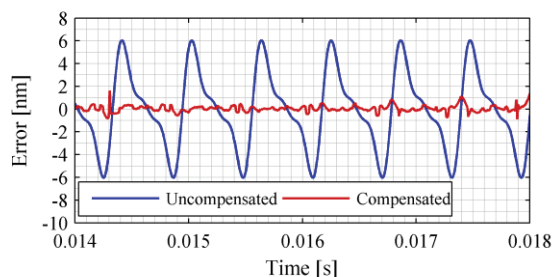


FIGURE 6. *The result of periodic error compensation in the non-constant velocity motion.*

CONCLUSIONS

A CWT-based algorithm is improved and used in this research to compensate higher order periodic error in heterodyne interferometer displacement signals for constant and non-constant velocity motions. Future work will focus on implementing this algorithm on the hardware and compensating the experiment-collected data in real-time.

ACKNOWLEDGEMENT

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REFERENCES

- [1] Fedotova G. Analysis of the measurement error of the parameters of mechanical vibrations. *Measurement Techniques*. 1980; 23: 577-580.
- [2] Quenelle R. Nonlinearity in interferometric measurements. *Hewlett-Packard Journal*. 1983; 34: 10.
- [3] Sutton C. Non-linearity in length measurement using heterodyne laser Michelson interferometry. *Journal of Physics E: Scientific Instruments*. 1987; 20: 1290-1292.
- [4] Badami V, Patterson S. A frequency domain method for the measurement of nonlinearity in heterodyne interferometry. *Precision Engineering*. 2000; 24: 41-49.
- [5] Schmitz T, Chu D, Kim H. First and second order periodic error measurement for non-constant velocity motions. *Precision Engineering*. 2009; 33: 353-361.
- [6] Ellis J, Baas M, Joo K, Spronck J. Theoretical analysis of errors in correction algorithms for periodic nonlinearity in

displacement measuring interferometers. *Precision Engineering*. 2012; 36: 261-269.

- [7] Lu C, Troutman J, Ellis J, Schmitz T, Tarbutton J. Periodic error compensation using frequency measurement with continuous wavelet transform. In *Proceedings of 29th ASPE*. 2014.
- [8] Daubechies I. The wavelet transform time-frequency localization and signal analysis. *IEEE Transactions on Information Theory*. 1990; 36: 961-1004.
- [9] Farge M. Wavelet transforms and their applications to turbulence. *Annual Review of Fluid Mechanics*. 1992; 24: 395-457.
- [10] Kaiser G. A friendly guide to wavelets. Birkhauser. 1994; 300pp.
- [11] Liu H, Cartwright AN, Basaran C. Moire interferogram phase extraction: a ridge detection algorithm for continuous wavelet transforms. *Applied Optics*. 2004; 15: 850-857.
- [12] Cherbuliez M, Jacquot P. Phase computation through wavelet analysis: yesterday and nowadays. *Fringe*. 2001: 154-162.

MODULAR VERSATILE IMAGING, FOCUSING & ILLUMINATION FOR VISION WITH HIGH STABILITY AND ACCURACY

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INTRODUCTION

Technological progress in the application, functionality and performance of Vision systems is tremendously. Especially in the field of image acquisition and image processing, this results in a growth in intelligent and smart systems. To support the increasingly demanding requirements for straightforward and robust integration of vision systems in machinery and equipment, SSvA developed a modular versatile imaging system for applications with high magnification levels (400x and higher). The autofocus feature creates a convenient sharp image, even when high magnifying objective lenses are used. By combining functional modules with a compact base body and standardized compatible interfaces, configurations for a wide range of vision applications can be made in a flexible and cost efficient manner.

The iMGR-2100 modular imaging system meets the challenges in accurate and consistent, stable image quality.

SYSTEM MODULES

The iMGR-2100 system can be configured by combination or customization of the different system modules.

Base Body Module

The heart of the iMGR-2100 system is the base body module. This monolithical, compact and robust body enables stable and reproducible image acquisition, even under demanding environmental and operational conditions.

The body module is equipped with interfaces for the different system modules and machine mounting. Multiple machine mounting interface possibilities enable the best suitable orientation (horizontally or vertically) of the system.

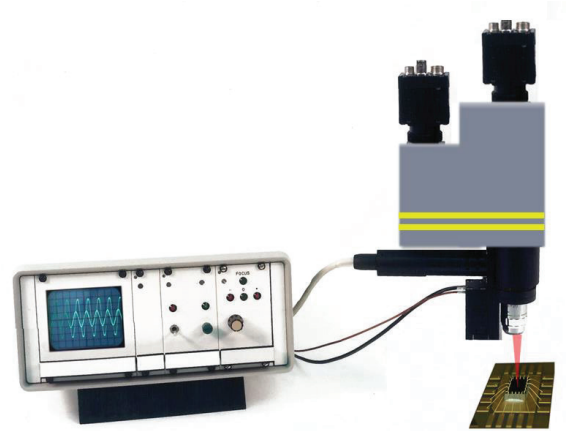


FIGURE 1. Overview of the iMGR-2100 system

Optical Modules

For the imaging of objects, micro- and nano-structures and assemblies, surfaces textures, etc., a magnification level of $> 400\times$ is required. The magnification factor can be configured by the selection of commercially available objective lenses, using a standard microscope interface. Simultaneous imaging at different magnifications is possible by configuring up to 2 optical camera channels. This enables the possibility to create an overview in combination with an image at the location of interest with higher magnification.

The camera channel is equipped with mounts for filters, polarizers, etc. for enhancement of the image quality. Various camera types and brands can be selected.

Auto-focus Module

Instant and continuous sharp and focused images can be acquired using the auto-focus module. Inspection of small details of an object requires highly magnifying imaging optics, which have a very small depth of focus. Manual or fixed focusing is not an ideal solution for these cases. In the auto-focus module, a piezo-driven stage positions the objective lens at the best focus position. The piezo-drive stage enables fast, stable and accurate manipulation over a 100 or 200 micrometer range, with an accuracy of less than 15 nm.

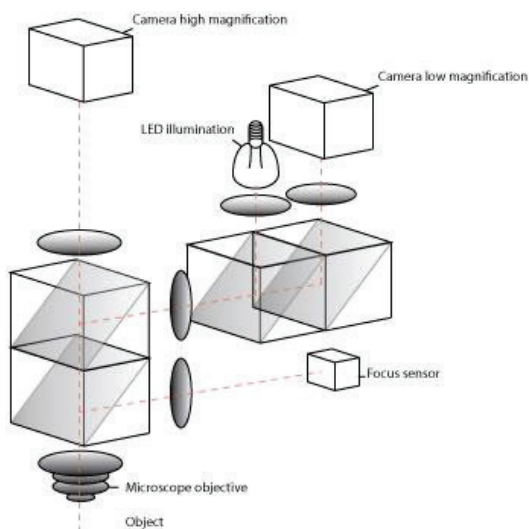


FIGURE 2. Configuration of the system

The best focus position of the acquired image is determined and maintained by an optical focus sensor, featuring a laser spot size of $\sim 2\ \mu\text{m}$, and closed-loop feedback to the piezo actuator.

The absolute focus position can be monitored or used as a height measurement with nm-range accuracy. The laser spot can be visualized in the camera image, indicating the actual measurement and focusing position in the image. This functionality enables the system to be used as an optical and contactless measurement tool with nm-range accuracy. In fact, this combination of the optical laser displacement sensor with integrated camera image (viewfinder and navigation), is one of the most powerful applications of the iMGR-2100 system.

The auto-focus module design is a further development based on the optical principles as used for optical disc read-out. Besides the application as focus sensor in the camera microscope, it can also be used as stand-alone optical distance measurement device. The focus sensor can be configured for a range of measurement ranges, free working distances and mechanical form-factor, to match the requirements of the application.

Illumination Module

The standard illumination module consists of a white LED source with adjustable optical output power. The module creates a homogenous intensity profile in the image area.

The illumination module can be customized for dedicated image analysis methods, for example by laser projection of patterns or line grids.

CONCLUSION

A modular versatile auto-focus camera microscope system is developed to support the vision applications that require highly magnified, stable and reproducible images. The optical focus sensor module enables automatic, fast and accurate focusing of the microscope image. The compact, robust and versatile system form factor makes it possible to integrate the system in many machine and equipment configurations. The focus sensor can also be used as a stand-alone system.

QUESTIONS

Please direct any questions to Mr. Gerrit van der Straaten at Settels Savenije USA, located in Fremont, CA. His contact information is listed below.

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REAL-TIME PERIODIC ERROR CORRECTION FOR HETERODYNE INTERFEROMETERS USING KALMAN FILTERS

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INSTRUCTIONS

Displacement measuring interferometry is a widely used technique for displacement metrology. For heterodyne interferometers, the metrology principle is that target position changes are recorded as a measured phase change. The relationship between the displacement and the measured phase is

$$x = \frac{\lambda}{4\pi n} \phi, \quad (1)$$

where x is the displacement of the target, N is the interferometer fold factor (two for the interferometer used in this work), n is the refractive index along the optical path difference, λ is the nominal wavelength of the laser light, and ϕ is the phase difference between the reference and measurement signals.

However, due to non-ideal laser sources, imperfect optical and electrical components, and misalignment, heterodyne interferometers suffer polarization mixing, frequency mixing, spurious ghost reflections, and unequal gain of detectors, which lead to a periodic error on the order of nanometers as a function of phase change (displacement) [1, 2, 3].

The measurement and reference signals influenced by above factors can be expressed by [4, 5]

$$I_m \propto \cos(2\pi f_s t + \phi) + \Gamma_{11} \cos(2\pi f_s t) + \Gamma_{12} \sin(2\pi f_s t) + \Gamma_{21} \cos(2\pi f_s t - \phi) + \Gamma_{22} \sin(2\pi f_s t - \phi), \quad (2)$$

$$I_r \propto \cos(2\pi f_s t), \quad (3)$$

where in (2), f_s is split frequency, ϕ is the phase difference between measurement and reference signals; the first term is nominal measurement signal, the second and third terms are the first-order periodic error, whose period is one cycle per fringe, and the fourth and fifth terms are second-order periodic error, whose period is two

cycles per fringe. Amplitudes Γ_{11} , Γ_{12} , Γ_{21} and Γ_{22} are corresponding coefficients of the periodic error. Typically, Γ_{11} and Γ_{21} contribute major periodic error, and value of Γ_{12} and Γ_{22} are very small, which are introduced by slight phase drift.

To extract the phase difference ϕ , lock-in detection and single-bin discrete Fourier transform methods are commonly used. The two signals (2,3) after multiplication and a low-pass filter can be expressed,

$$I_x \propto \cos(\phi) + \Gamma_{11} + \Gamma_{21} \cos(-\phi) + \Gamma_{22} \sin(-\phi), \quad (4)$$

$$I_y \propto \sin(\phi) - \Gamma_{12} + \Gamma_{21} \sin(-\phi) - \Gamma_{22} \cos(-\phi). \quad (5)$$

They can be expressed in a generic form [5],

$$I_x = a \cos(\phi + \varphi) + I_{xc}, \quad (6)$$

$$I_y = b \sin(\phi) + I_{yc}, \quad (7)$$

where I_{xc} and I_{yc} are the DC offsets, a and b are the gain of the AC components, and φ is the quadrature phase drift of ϕ (not exactly $\pi/2$ between I_x and I_y). The in-phase I_x and quadrature I_y signals for homodyne interferometers have the same expression [6], so the following method works for both for heterodyne and homodyne interferometers.

In ideal setup, $a = b$, $I_{xc} = I_{yc} = 0$ and $\varphi = 0$, where the phase difference ϕ could be exacted by an arctangent operation exactly:

$$\phi = \arctan \left(\frac{\sin(\phi)}{\cos(\phi)} \right). \quad (8)$$

However, in practice, the DC offsets and quadrature phase shift are non-zero, the gains are unequal, so the phase is also affected by the periodic error. The phase actually should be extracted by (9)

$$\phi = \arctan \left(\frac{I_y - I_{yc}}{(I_x - I_{xc}) \cdot \alpha + (I_y - I_{yc}) \cdot \beta} \right), \quad (9)$$

where α and β are correction coefficients in terms of a , b , and φ .

$$\alpha = \frac{b}{a \cos(\varphi)}, \quad (10)$$

$$\beta = \tan(\varphi). \quad (11)$$

There are two ways to deal with periodic error to achieve more precise measurement, one is to design an advanced optical and mechanical configuration of interferometer, which is inherently free from periodic error [7, 8]. Another is to develop a compensation algorithm applied on measurement data to reduce the periodic error electronically [9, 10].

Several algorithms have been proposed to correct the periodic error. Among them, ellipse fitting is an essential correction method in interferometer applications. Heydemann [9] presents a method to identify and correct the gain ratio b/a , DC offsets and quadrature phase shift in (6,7) by fitting the ellipse, then calculate the phase by using the corrected reference and measurement signals. Birch [10] improves the method and uses a computer to simulate this correction method.

In this paper, we investigate and improve the implementation of ellipse fitting method.

ELLIPSE FITTING

Mathematically, the Lissajous curve of (I_x, I_y) is an ellipse (Figure 1). In analytic geometry, the point (I_x, I_y) satisfies

$$AI_x^2 + BI_xI_y + CI_y^2 + DI_x + EI_y + F = 0, \quad (12)$$

where the coefficients (A, B, C, D, E, F) determines the ellipse. By estimating the ellipse coefficients, the correction coefficients α , β , I_{xc} and I_{yc} can be calculated by (A, B, C, D, E, F) . Since the trace $A + C$ can never be zero for an ellipse, the arbitrary scale factor can be removed by the normalization $A + C = 1$. Thus, the ellipse can be described by a vector (A, B, D, E, F) [11].

Estimating the ellipse parameters from noisy data is a common problem in the field of computer vision. Several techniques for ellipse fitting have been presented, such as linear least-squares, orthogonal least-squares, gradient-weighted least-squares; bias-corrected renormalization; Kalman filter and robust techniques [12].

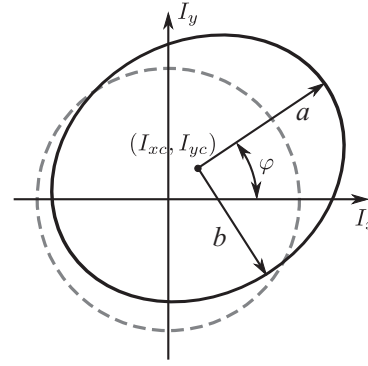


FIGURE 1. The Lissajous curve of raw (I_x, I_y) (ellipse) and corrected (I_x, I_y) (circle). For those coefficients in (6,7), I_{xc} and I_{yc} are the center of ellipse, a and b are major and minor semi-axes, and φ is the angle between the X-axis and the major axis. The goal of ellipse fitting is to eliminate the influence of DC offset, unequal gain and quadrature phase shift.

Some of them have been adjusted and applied in periodic error correction, including linear least-squares and Kalman filter. Wu, *et al.* [6] apply a linear least-squares method on a set of measurement data (I_x, I_y) . It uses a computer to do post-processing for all fitting and correction operations. Thus, the corrected data is not suitable for further real-time measurement and control use. Eom, *et al.* [13] present an implementation to adjust (I_x, I_y) in real time by using a tunable analog circuit. However, the correction coefficients still must be calculated on a computer using preliminary measurement data. Also, correction coefficients must be treated as constant during one measurement period. Furthermore, the analog circuit is suspected of introducing other noise sources.

Chawah, *et al.* [14] set up a system to estimate coefficients and correct the error based on a Kalman filter on a computer. However, the measurement data has to transfer from the data acquisition system (FPGA) to the computer, and the corrected data is difficult to use in future real-time measurement and control.

Kim, *et al.* [15] develop a digital signal processing module for real-time periodic error compensation. It continuously obtains specific phase result at the multiples of $\pi/4$ to estimate the correction coefficients through simple arithmetic calculations, instead of matrix operations. However, it is easily affected by the accuracy of those specific phases

and the speed of the phase change.

From the above investigations, a linear least-square method usually leads to a closed-form solution, however, it must wait for all measurement data to be available, and then operate a single joint evaluation, which is a set of huge matrix operations and impractical to run in real-time. Unlike a linear least-square method, a Kalman filter has a recursive nature, which makes it is suitable for processing data in a time series [12]. Meanwhile, once meeting the tolerance, it could exit the loop immediately. So it needs far less time to achieve a satisfactory result.

In this paper, we present an approach to estimate and update correction coefficients online and correct the periodic error in real time based on a Kalman filter, which is capable to of correcting time-varying periodic error, and develop a compact phasemeter system integrating data acquisition, coefficients estimation, phase demodulation and periodic error correction all in one FPGA board.

EXTENDED KALMAN FILTER

Kalman filter gives the optimal estimation of a linear system model. However, in practice, most system models are nonlinear, including the expression of an ellipse. An extended Kalman filter (EKF) linearizes the model at each point using a Taylor series expansion (like a tangent line at a certain point on a curve). Then the EKF could estimate the system model using the methodology of Kalman filter [12].

In this design, EKF is applied to estimate the ellipse coefficients (A, B, D, E, F) . Let estimation $\mathbf{x}$ equal vector $(A, B, D, E, F)^T$, it can be recursively computed using [11],

$$\mathbf{z}_k = -F(\mathbf{x}, I_x, I_y), \quad (13)$$

$$\mathbf{K}_k = \mathbf{P}_{k-1|k-1} \mathbf{H}_k^T / (\mathbf{H}_k \mathbf{P}_{k-1|k-1} \mathbf{H}_k^T + \mathbf{R}_k), \quad (14)$$

$$\hat{\mathbf{x}}_{k|k} = \hat{\mathbf{x}}_{k-1|k-1} + \mathbf{K}_k \mathbf{z}_k, \quad (15)$$

$$\mathbf{P}_{k|k} = (\mathbf{I} - \mathbf{K}_k \mathbf{H}_k) \mathbf{P}_{k-1|k-1}, \quad (16)$$

where $\mathbf{z}_k$ is observation of true state $\mathbf{x}_k$, $F(\mathbf{x}, I_x, I_y)$ is the left side of (12); $\mathbf{K}_k$ is optimal Kalman gain, $\mathbf{H}_k$ (17) is observation model, $\mathbf{R}_k$ (18) is covariance of observation noise; $\hat{\mathbf{x}}_k$ is the estimated ellipse coefficients; $\mathbf{P}_k$ is the predicted estimate covariance matrix, whose initial value is identity matrix.

Equations (13-16) are updated in each iteration.

After certain number of iterations, the ellipse coefficients $\mathbf{x}$ converge to constant value, then they can be used for correcting periodic error. Because ellipse model is nonlinear, in order to linearize the model, observation model $\mathbf{H}_k$ and covariance of observation noise $\mathbf{R}_k$ have to be updated in each iteration too,

$$\mathbf{H}_k = \nabla_{\mathbf{x}} F(\mathbf{x}, I_x, I_y), \quad (17)$$

$$\mathbf{R}_k = \Sigma^2 ((\nabla_{I_x} F)^2 + (\nabla_{I_y} F)^2), \quad (18)$$

where Σ is the noise level.

IMPLEMENTATION

The implementation of this module consists of two parts: one is correction coefficient estimation and another is periodic error correction.

The first part is to implement a Kalman filter to estimate ellipse coefficients and calculate correction coefficients in hardware. Two solutions are commonly used to implement the Kalman filter in hardware. One is to map it to an HDL-designed, parallel, pipelined architecture in FPGA, also known as systolic array [16]. This solution inherently has high throughput, which means it could update those four correction coefficients at high frequency (once in several clock cycles). However, to design the systolic array architecture is difficult. Another is to program Kalman filter algorithm in C/C++, and execute it in a microprocessor, which is straightforward for design and rapid for algorithm verification and prototyping. Due to the occurrence of matrix multiplications, which consist of an amount of basic arithmetic operations, the microprocessor must execute each basic operations for each input data serially, which leads to low throughput (thousands of clock cycles).

For this proceedings, we chose the second solution. To integrate it with our existing FPGA-based phasemeter, we use an Altera Nios II embedded processor to manage each step of the Kalman filter, and use a co-processor to execute basic floating-point arithmetic operations. The schematic of the module is shown in Figure 2.

Due the limitation of maximum clock frequency of the soft processor and complicated matrix operations of Kalman filter, the Kalman filter running in the soft-processor cannot update the correction coefficients for every (I_x, I_y) , which is generated at 50 MHz. Some attempts have been done to increase the updating frequency as

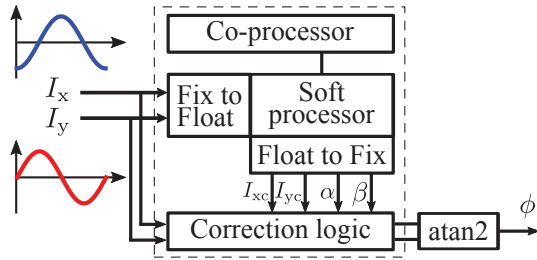


FIGURE 2. The periodic error correction module consists of two main components. One is an embedded soft processor with a floating-point arithmetic co-processor in FPGA to implement Kalman filter algorithm and update correction coefficients. Another is a dedicated logic circuit to correct the periodic error by using those correction coefficients. After correction, an arctangent (atan2) operation is used to extract the phase.

fast as possible: increasing the clock frequency of soft-processor to its upper limit 100 MHz (if SOC FPGA is available, the ARM processor on it could run at 800 MHz); using single-precision floating-point data type, instead of double-precision, it increases the updating frequency about 30% and still keeps enough precision (fixed-point data type may be even faster, which is supported by the ARM processor); using floating-point arithmetic co-processor, which provides dedicated logic circuit to execute the basic arithmetic operations, instead of executing them in processor, it increases the updating frequency at least two times; applying software optimization and hardware architecture simplification.

Currently, this module updates correction coefficients at 6.5 kHz. Once the interferometer configuration setup, the correction coefficients changes very slowly, so the updating frequency is sufficient for identifying the correction coefficients of a time-varying periodic error, provided the coefficients change slower than ~ 1 kHz. Meanwhile, it estimates the ellipse coefficients online automatically, avoids to transfer measurement data to a computer for offline Kalman filter and go back to set correction coefficients.

Besides the updating frequency, the convergence time is also important for ellipse fitting. By setting initial value of estimated ellipse coefficients $\mathbf{x}$ as (0.5, 0, 0, 0, -0.125) (unit circle) and the noise level Σ as 0.05, it could only use as few as 10 points in one period to estimate optimal enough ellipse coefficients.

The second part corrects the periodic error based on the output of the Kalman filter. For some applications, they must compensate I_x and I_y signals with equal gains, zero DC offsets and zero quadrature phase shift, so a, b, φ, I_{xc} and I_{yc} must be calculated in terms of the ellipse coefficients. For our phasemeter, we are more interested in the phase ϕ , we could compensate I_x and I_y first and then calculate ϕ using (8), but a much more straightforward way is correcting the error using (9) directly, in which the coefficients I_{xc}, I_{yc}, α and β is expressed in terms of (A, B, C, D, E, F) [9]:

$$I_{xc} = \frac{2CD - BE}{B^2 - 4AC}, \quad (19)$$

$$I_{yc} = \frac{2AE - BD}{B^2 - 4AC}, \quad (20)$$

$$\alpha = \frac{2A}{\sqrt{4AC - B^2}}, \quad (21)$$

$$\beta = \frac{B}{\sqrt{4AC - B^2}}. \quad (22)$$

The correction logic calculates $I_y - I_{yc}$ and $(I_x - I_{xc}) \cdot \alpha + (I_y - I_{yc}) \cdot \beta$. For FPGA, it is difficult to implement fixed-point division, so a CORDIC-based arctan2 is used, which accepts the numerator and denominator (9) separately, thus, avoids the division operation.

Although the correction coefficients are not updated for every input (I_x, I_y), the correction logic can correct each output phase in real time. Hence, the corrected phase can be used in future measurement and control steps.

SIMULATIONS

Simulations have done to verify the functionality and performance of this periodic error correction module. A pair of generated reference (I_r) and measurement (I_m) signals with coefficients ($\Gamma_{11}, \Gamma_{12}, \Gamma_{21}, \Gamma_{22}$) are fed into the phasemeter with this module running on the FPGA and a set of correction coefficients are recursively computed.

If ($\Gamma_{11}, \Gamma_{12}, \Gamma_{21}, \Gamma_{22}$) = (0.08, 0.03, 0.05, 0.02), the Kalman filter only needs 10 (I_x, I_y) samples on the ellipse and 1.45 ms time to converge the ellipse coefficients to a constant value. Figure 3 shows the value of the ellipse coefficients after each iteration.

Using the ellipse coefficients, the soft processor calculates the correction coefficients $I_{xc}, I_{yc}, \alpha, \beta$ and then correction logic and atan2 extract the corrected phase. Figure 4 shows the

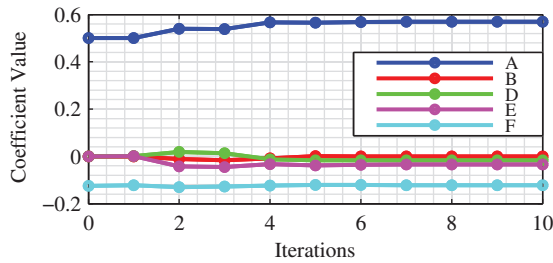


FIGURE 3. The ellipse coefficients converge to constant values after 10 times iterations. The initial values of ellipse coefficients are (0.5,0,0,0,-0.125). For a fast convergence, these 10 samples should approximately equally distribute on one cycle. If the periodic error is time-varying, after first convergence, the coefficients can be gradually adjusted by future iterations.

performance of this periodic error correction module.

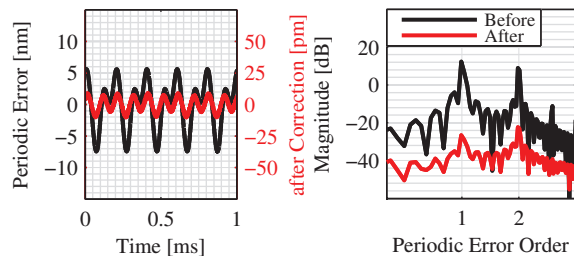


FIGURE 4. Comparison of periodic error before and after correction. The left figure shows in the time domain, a total ± 7.5 nm periodic error attenuates to ± 10 pm. The right figures shows in terms of error order, the first- and second- order error attenuate 38.9 dB and 31.4 dB, respectively.

CONCLUSION

We present a periodic error correction module for heterodyne interferometer, and compatible with homodyne interferometer. This module is implemented on FPGA, which can online estimate ellipse coefficients using extended Kalman filter, update the correction coefficients at 6.5 kHz, and correct the periodic error in real time. It has capability to deal with slow time-varying periodic error.

FUTURE WORK

We will attempt to achieve faster correction coefficients updating speed, which currently is the bottleneck of this model. The Kalman filter may need to be implemented in an ARM processor

of a SOC chip, which is a hard processor, can run much faster than soft processor and support fixed-point operations. Or map the Kalman filter to a systolic array, which is pure logic circuit, could run even higher frequency and have higher throughput.

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REFERENCES

- [1] Quenelle RC. Nonlinearity in interferometer measurements. *Hewlett-Packard Journal*. 1983;34:10.
- [2] Rosenbluth AE, Bobroff N. Optical sources of non-linearity in heterodyne interferometers. *Precision Engineering*. 1990;12(1):7–11.
- [3] Bobroff N. Recent advances in displacement measuring interferometry. *Measurement Science and Technology*. 1993;4(9):907.
- [4] Wu CM, Deslattes RD. Analytical modeling of the periodic nonlinearity in heterodyne interferometry. *Appl Opt*. 1998 Oct;37(28):6696–6700.
- [5] Eom T, Choi T, Lee K, Choi H, Lee S. A simple method for the compensation of the nonlinearity in the heterodyne interferometer. *Measurement Science and Technology*. 2002;13(2):222.
- [6] Wu CM, Su CS, Peng GS. Correction of nonlinearity in one-frequency optical interferometry. *Measurement Science and Technology*. 1996;7(4):520.
- [7] Joo KN, Ellis JD, Buice ES, Spronck JW, Schmidt RHM. High resolution heterodyne interferometer without detectable periodic nonlinearity. *Opt Express*. 2010 Jan;18(2):1159–1165.
- [8] Ellis JD, Meskers AJH, Spronck JW, Schmidt RHM. Fiber-coupled displacement interferometry without periodic nonlinearity. *Opt Lett*. 2011 Sep;36(18):3584–3586.

- [9] Heydemann PLM. Determination and correction of quadrature fringe measurement errors in interferometers. *Appl Opt.* 1981 Oct;20(19):3382–3384.
- [10] Birch KP. Optical fringe subdivision with nanometric accuracy. *Precision Engineering.* 1990;12(4):195–198.
- [11] Porrill J. Fitting Ellipses and Predicting Confidence Envelopes Using a Bias Corrected Kalman Filter. *Image Vision Comput.* 1990 Feb;8(1):37–41.
- [12] Zhang Z. Parameter estimation techniques: a tutorial with application to conic fitting. *Image and Vision Computing.* 1997;15(1):59–76.
- [13] Eom TB, Kim JA, Kang CS, Park BC, Kim JW. A simple phase-encoding electronics for reducing the nonlinearity error of a heterodyne interferometer. *Measurement Science and Technology.* 2008;19(7):075302.
- [14] Chawah P, Sourice A, Plantier G, Chery J. Real time and adaptive Kalman filter for joint nanometric displacement estimation, parameters tracking and drift correction of EFFPI sensor systems. In: *Sensors, 2011 IEEE*; 2011. p. 882–885.
- [15] Kim JA, Kim JW, Kang CS, Eom TB, Ahn J. A digital signal processing module for real-time compensation of nonlinearity in a homodyne interferometer using a field-programmable gate array. *Measurement Science and Technology.* 2009;20(1):017003.
- [16] Bigdeli A, Biglari-Abhari M, Salcic Z, Lai YT. A New Pipelined Systolic Array-based Architecture for Matrix Inversion in FPGAS with Kalman Filter Case Study. *EURASIP J Appl Signal Process.* 2006 Jan;2006:75–75.

COMPACT SIX DEGREE-OF-FREEDOM INTERFEROMETER FOR PRECISION LINEAR STAGE METROLOGY

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INTRODUCTION

Precision linear stages play an invaluable role in industrial manufacturing and measuring machines. The ultimate performance of the equipment depends largely on the linear stage movement accuracy. There are six geometric motion errors associated with a precision linear stage. The main movement along z-axis is the displacement, Δz ; there are two straightness errors, Δx and Δy , and three rotational errors, θ_x (pitch), θ_y (yaw), and θ_z (roll).

Current commercial interferometers utilize various components and accessories to measure different degrees of freedom (DOFs) and most of them can only measure one DOF at a time. In this case, several setups are needed in the calibration process for a precision linear

stage. Furthermore, extra errors might be generated from different system alignments and environmental parameters in each measurement. To solve this problem, many metrology systems have been proposed to realize simultaneous measurement of multiple degrees of freedom. Some are based on laser interference and autocollimation [1-3] and some are based on Michelson interferometry and grating shear [4]. Even though these systems have the ability to measure six DOFs simultaneously, the bulky size, limited measurement resolution, and/or limited range are significant shortcomings.

In this proceeding, we propose a compact fiber delivered 6-DOF interferometer based on differential wavefront sensing, selective laser polarization and autocollimation.

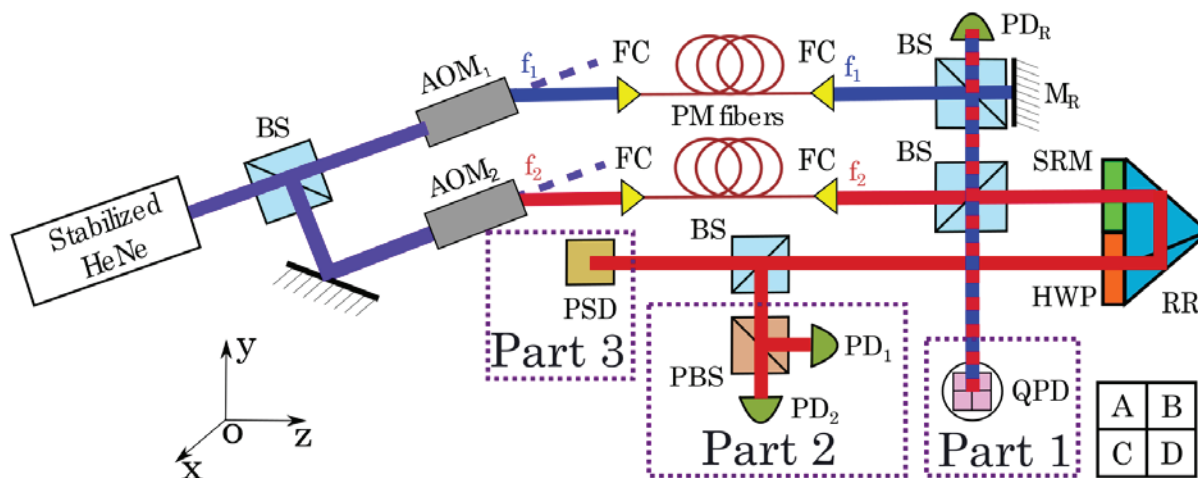


FIGURE 1. Schematic of the 6 DOF interferometer. (BS: beamsplitter, AOM: acousto-optic modulator, FC: fiber coupler, PD: photodetector, M: mirror, SRM: semi-reflective mirror, HWP: half-wave plate, RR: retro-reflector, QPD: quadrant photodetector, PBS: polarized beamsplitter, PSD: position sensitive detector)

WORKING PRINCIPLE

The schematic of our 6-DOF interferometer is shown in figure 1. The light from a stabilized laser source is split equally and directed towards two acoustic-optic modulators (AOMs), which are driven at slightly different RF frequencies to create a frequency difference between the two beams. After the AOMs, the first orders of the two beams are coupled into polarization maintaining fibers and then collimated back into free space. The f_1 beam passes through the top non-polarizing beamsplitter (BS), reflects from the fixed reference mirror and interferes with the f_2 beam that is reflected from the bottom BS at reference detector PD_R. Likewise, the f_2 beam passes through the bottom BS, half of the beam reflects from the retro-reflector with a semi-reflective coating part, and interferes back at the bottom BS with the f_1 beam that is reflected from the top BS at measurement quadrant photodetector (QPD). Meanwhile, the other half of the f_2 beam transmits through the semi-reflective coating, reflects from the retro-reflector and then transmits through the half-wave plate (HWP) which is fixed with the retro-reflector. Afterwards, half of the beam is incident on the position sensitive detector (PSD), the other half is split by a polarizing beamsplitter (PBS) and the optical power in orthogonal polarization directions is measured by PD₁ and PD₂.

DISPLACEMENT, PITCH AND YAW

Part 1 adopts the differential wavefront sensing (DWS) technique that utilizes a quadrant photodetector as the measurement detector to realize simultaneous measurements of displacement (Δz), pitch (θ_x), and yaw (θ_y). The individual phase change at each quadrant (A, B, C and D) relative to the reference photodetector is detected and computed from the interference wavefront. The displacement is determined by averaging the measured phase between all quadrants. Pitch and yaw is determined by creating a weighted phase average over symmetrically adjacent quadrant detector pairs. The three DOFs are decoupled from DWS signals using,

$$z \propto \frac{\phi_A + \phi_B + \phi_C + \phi_D}{4}, \quad (1)$$

$$pitch \propto \frac{(\phi_A + \phi_B) - (\phi_C + \phi_D)}{L_p}, \text{ and} \quad (2)$$

$$yaw \propto \frac{(\phi_A + \phi_C) - (\phi_B + \phi_D)}{L_y}, \quad (3)$$

where ϕ represents detected phase of quadrant A, B, C or D as shown in figure 1, and L_p and L_y represent an equivalent length in pitch and yaw measurements which is primarily dependent on beam diameter, detector size, alignment errors and beam wavefront. A more detailed description for this technique has been published in [5] and we are currently developing a numerical model to give a better estimation for the DWS readout. Figure 2 shows our yaw measurement result compared with a Renishaw angle interferometer measuring from the opposite direction for a linear stage motion within a 15 mm range, the standard deviation of error is 0.3 μ rad. A noise floor measurement for displacement, pitch and yaw over 80 seconds is shown in figure 3, which indicates our 6 DOF interferometer can achieve 1 nm displacement resolution and 0.4 μ rad in pitch and yaw.

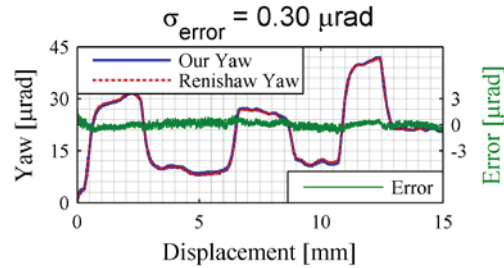


FIGURE 2. The comparison of yaw measurement with a Renishaw angle interferometer during linear motion within 15 mm.

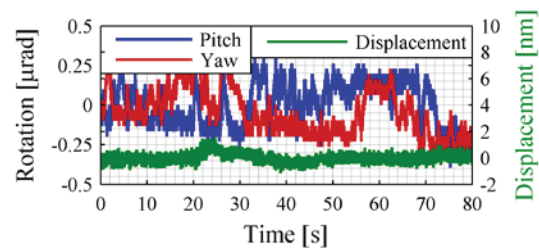


FIGURE 3. The noise floor measurements for pitch, yaw and displacement over 80 seconds using DWS proposed for our 6 DOF interferometer.

ROLL

Part 2 uses two photodetectors to measure the optical power in orthogonal polarization directions, which contains roll (θ_z) information. The original polarization state for the f_2 beam is

45 degrees and the mathematical expression is the electric field. For PD₁ and PD₂, they measure the optical power component E_x in the x-direction and E_y in the y-direction, respectively. Thus, when the retro-reflector target has a roll angle change θ the measured optical power in PD₁ and PD₂ can be express as,

$$V_1 = E_x^2 = E^2 \cos^2(2\theta + 45^\circ) = \frac{E}{2}(1 - \sin 4\theta), \quad (4)$$

$$V_2 = E_y^2 = E^2 \sin^2(2\theta + 45^\circ) = \frac{E}{2}(1 + \sin 4\theta). \quad (5)$$

Therefore, the roll information can be calculated as,

$$\theta = \frac{1}{4} \arcsin\left(\frac{V_1 - V_2}{V_1 + V_2}\right). \quad (6)$$

Currently, we keep constant amplitude of the f_2 beam. It will be better to modulate the amplitude as a square wave by using a transistor-transistor logic (TTL) input from a function generator as described in [6], which can cancel common mode noise and achieve higher resolution. Figure 4 shows our single roll measurement compared with a Renishaw angle interferometer pitch measurement from the orthogonal direction. The standard deviation of the error over a long term measurement is 13.8 μrad .

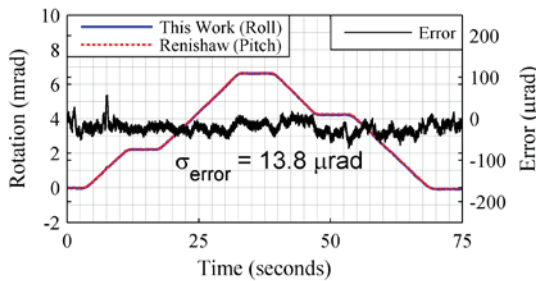


FIGURE 4. The roll measurement compared with a Renishaw angle interferometer pitch measurement from the orthogonal direction.

In our preliminary roll measurement experiments using a half-wave plate (HWP), we noticed that the roll information is coupled with pitch and yaw motions. Even to some extent, the final result can be corrected using the pitch and yaw information as a feedback mechanism. Replacing the HWP with a polarizer might be one solution and the mathematical relation is as follows.

$$V_1 = E_x^2 = E^2 \cos^2 \theta \cos^2(\theta + 45^\circ) \quad (7)$$

$$V_2 = E_y^2 = E^2 \cos^2 \theta \sin^2(\theta + 45^\circ) \quad (8)$$

Thus, the roll information can be determined by,

$$\theta = \frac{1}{2} \arcsin\left(\frac{V_1 - V_2}{V_1 + V_2}\right). \quad (9)$$

STRAIGHTNESS

Part 3 utilizes a position sensitive detector to determine the straightness (Δx and Δy) in two directions. Rather than using the quadrant photodetector in [7], we replace it with a position sensitive detector (PSD) which gives a voltage readout that is proportional to the centroid position of incident beam. The PSD has two normalized power outputs from -10 to 10 volts which correspond to beam center displacements from -10 to 10 mm across the entire work volume. Thus, the straightness errors can be determined by measuring the beam centroid change Δx_{PSD} and Δy_{PSD} on the PSD as,

$$\Delta x = \frac{\Delta x_{PSD}}{2}, \quad (10)$$

$$\Delta y = \frac{\Delta y_{PSD}}{2}. \quad (11)$$

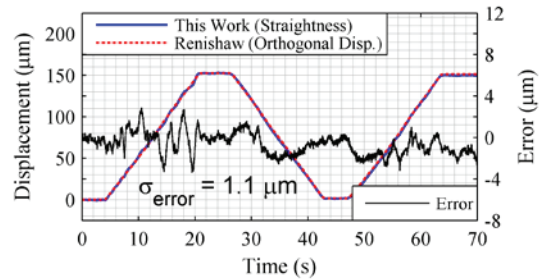


FIGURE 5. The single straightness error measurement compare with Renishaw displacement measurement in orthogonal direction.

In practice, the coefficient will not exactly be 1/2 and a calibration factor is needed to achieve better accuracy. Figure 5 shows the comparison between our straightness measurement with a Renishaw displacement measurement from orthogonal directions over a 150 μm range. The standard deviation of the error is 1.1 μm and the resolution is 1 μm in both straightness directions.

CONCLUSION

A compact 6-DOF interferometer capable of simultaneously measuring displacement, pitch, yaw, straightness and roll for precision linear stages has been presented in this proceeding. The working principle for measuring each DOF has been described and the resolution has been given. Some experiments have been done to validate single DOF measurement results. We are currently working on combining different measurement parts together to realize a full 6-DOF measurement comparison with a commercial metrology system for a precision linear stage calibration.

FUTURE WORK

The resolution for straightness measurement is still relatively large compared with the current state of the art. Thus, we are currently seeking other approaches for straightness measurement to further improve the resolution. In addition, the repeatability experiments will be performed and the cross-talk errors will be investigated for each DOF. Finally, we would like to have a system design for packaging all the optical components and electronics together in a relatively small footprint, which will be more stable and convenient for future use.

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REFERENCES

- [1] Lee, S.W., R. Mayor, and J. Ni, *Development of a six-degree-of-freedom geometric error measurement system for a meso-scale machine tool*. Journal of manufacturing science and engineering, 2005. **127**(4): p. 857-865.
- [2] Qibo, F., et al., *Development of a simple system for simultaneously measuring 6DOF geometric motion errors of a linear guide*. Optics Express, 2013. **21**(22): p. 25805-25819.
- [3] Chen, B., et al., *Laser straightness interferometer system with rotational error compensation and simultaneous measurement of six degrees of freedom error parameters*. Optics Express, 2015. **23**(7): p. 9052-9073.
- [4] Hsieh, H.-L. and S.-W. Pan, *Development of a grating-based interferometer for six-degree-of-freedom displacement and angle measurements*. Optics Express, 2015. **23**(3): p. 2451-2465.
- [5] Gillmer, S.R., et al., *Compact fiber-coupled three degree-of-freedom displacement interferometry for nanopositioning stage calibration*. Measurement Science and Technology, 2014. **25**(7): p. 075205.
- [6] Gillmer, S.R., et al., *Robust high-dynamic-range optical roll sensing*. Optics Letters, 2015. **40**(11): p. 2497-2500.
- [7] Feng, Q., B. Zhang, and C. Kuang, *A straightness measurement system using a single-mode fiber-coupled laser module*. Optics & Laser Technology, 2004. **36**(4): p. 279-283.

SURFACE FINISH AND 3D OPTICAL SCANNER MEASUREMENT PERFORMANCE FOR PRECISION ENGINEERING

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ABSTRACT

3D optical scanners are rapidly replacing Cartesian CMMs due to their advanced metrological benefits, such as dense 3D point cloud data capture, portability and measurement speed. To support 3D optical scanner manufacturers and end-users, the National Physical Laboratory (NPL, UK) has developed a 3D optical scanner dimensional characterisation facility, which comprises: test artefacts, test procedures and equipment to examine the performance of 3D optical scanners.

One of the facility's tests characterises scanner ability to measure surface finishes common to precision engineering. The test utilises a multi-faceted test artefact, to allow measurement of the test materials against highly Lambertian reference coupons through a series of fixed angles. Sample surfaces representative of materials typical of precision components manufactured by the aerospace, automotive and power generation industries were measured. Results for a white light fringe projecting 3D optical scanner show sensitivity to certain aluminium finishes and specular surfaces. Measurements of sandpaper samples suggest a trend of increasing point cloud density with decreasing surface roughness. Colour measurements show sensitivity to blue and green surfaces, while maximum point cloud density is measured for other coloured surfaces.

INTRODUCTION

3D optical surface-form scanners, such as fringe projectors and laser scanning articulating arms, have advanced metrology benefits that are particularly desirable on the workshop floor. While not offering Cartesian CMM levels of accuracy, their dense 3D point cloud data capture, portability and speed make them particularly attractive on precision component production lines; particularly in the aerospace, automotive and power generation industries.

CMMs are a well understood [1], trusted and established technology [2] that has matured since

their initial development in the 1950s [3]. Traceability from CMM measurements is achieved using a substitution method based on calibrated artefacts and virtual CMM techniques [4, 5]. In contrast, portable 3D optical scanners are a relatively new and advancing technology; international standards on their acceptance and use are yet to be published and techniques to provide measurement traceability are still being developed [6].

Recognising industry's need for a robust metrological infrastructure and traceability in 3D measurements, the National FreeForm Centre at the NPL has developed a 3D optical scanner dimensional characterisation facility [7]. The facility comprises a purpose-built environmentally-controlled laboratory (approximate dimensions: 3 m × 5 m × 2.5 m), test artefacts, test procedures and equipment to examine the performance of 3D optical surface-form scanners.

This paper reports measurements, from one of the test procedures within the facility, to verify performance when measuring surface finishes common to precision engineering. The aim of this work is to promote best practice when performing 3D optical scanner measurements and to develop an understanding of surface properties as a measurement uncertainty component.

SURFACE FINISH CHARACTERISATION ARTEFACT

The optical properties of an object's surface are known to affect 3D optical scanner measurements [8]. For example, while Lambertian surfaces that diffusely reflect projected light are ideal for 3D optical scanner measurements, little useful data may be recorded from specular surfaces as projected light is reflected away. Surface colour has also been shown to affect the data that a 3D optical scanner is able to record [9].

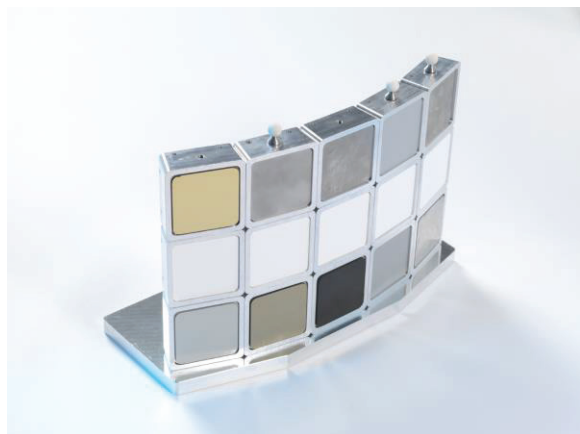


FIGURE 1: The NPL 3D material coupon plate.

The National FreeForm Centre has developed the NPL 3D material coupon plate, shown in figure 1. It consists of a multi-faceted test artefact to quantify 3D optical scanner's ability to measure different surface finishes (reflectance, roughness and colour).

NPL's 3D material coupon plate comprises three rows and five columns of 60 mm x 60 mm square coupons; a 150° angle separates each column rather than forming a continuous plane. An alignment jig allows rotation of the NPL 3D material coupon plate such that the coupons within each column can be measured at fixed positions normal to, at $\pm 10^\circ$ and $\pm 20^\circ$ angles relative to the scanner, as Figure 2 shows.

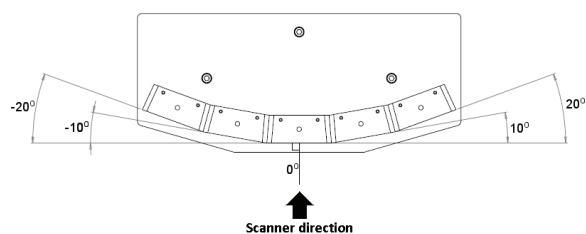


FIGURE 2: Top-down diagram of the NPL 3D material coupon plate.

Five highly Lambertian, sintered PTFE (Spectralon®) reference coupons are placed along the plate's central row, while the sample coupons are placed in the rows above and below. Sample materials investigated include carbon-fibre-reinforced polymer, steel foil, various aluminium finishes, colour (using RAL colour standards [10]) and surface roughness (using different grades of sandpaper).

SURFACE FINISH MEASUREMENTS

The National FreeForm Centre has three white light fringe projecting 3D optical scanners with similar measurement volumes: a Breuckmann stereoSCAN 3D-HE, a GOM ATOS IIe and Steinbichler Comet 5 1.4M. For the purposes of impartiality, the systems are referred to as "System A", "System B" and "System C", in no particular order.



FIGURE 3: Example point cloud of the NPL 3D material coupon plate produced by system A.

Measurements were performed with each column normal to, at $\pm 10^\circ$ and $\pm 20^\circ$ relative to the scanner, using optimal measurement parameters, as advised by the scanner's manufacturer. To account for measurement variance, six scans were performed at each coupon position and the results from the analysed point clouds averaged. Both positive and negative angles were chosen to test sensitivities for the non-symmetrical projector/camera arrangements of some systems, such as the Breuckmann and Steinbichler scanners. The point clouds from the measurements, an example of which is shown in Figure 3, were analysed using Polyworks 2014 software to identify a 30 mm by 30 mm region within the centre of each coupon. The number of points within this region was recorded and normalised against an identical region within the Spectralon reference coupon for the column. Thus, measuring data quantity and therefore determining a scanner's ability to measure a particular material's surface finish at a particular angle.

AEROSPACE MATERIAL MEASUREMENTS

Figure 4 presents results from System C showing scanner sensitivity to different aluminium surface finishes.

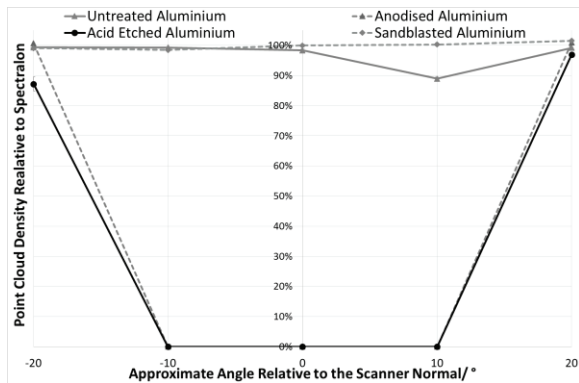


FIGURE 4 Point cloud densities for various aluminium finishes produced by System C normalised against the Spectralon reference.

The point cloud densities for the untreated and sandblasted aluminium coupons are roughly equal to their respective Spectralon references, meaning that System C had no difficulty in measuring them. The diffuse finishes of these samples are near ideal for these non-contact optical measurements. Samples with surface finishes that appear to be more specular, such as the acid etch and anodised aluminium coupons, were more difficult to measure, with no points recorded at normal and $\pm 10^\circ$ angles. However, at $\pm 20^\circ$ angles the point cloud densities are almost equal to their respective Spectralon references.

Figure 5 shows the relative point cloud densities that System C produced for other materials.

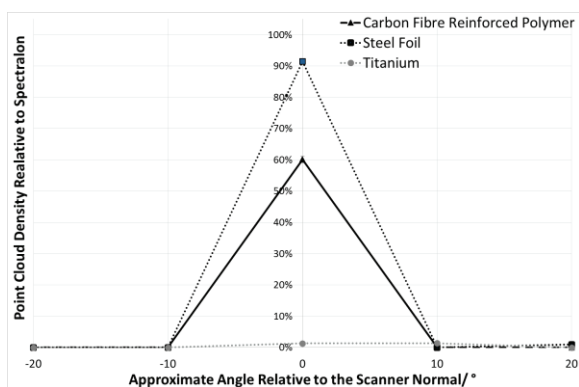


FIGURE 5 Normalised point-cloud densities for different aerospace materials produced by System C.

Specular reflection of the fringes projected onto the surface of the steel foil sample resulted in points being measured only when the coupon was normal to the scanner. The same was true for the black carbon fibre reinforced polymer sample. When normal to the scanner, the point cloud

density was approximately 60 % of the reference suggesting holes in the point cloud, which may affect dimensional measurements of components made from this material. Further investigation is required into which surface of carbon fibre reinforced polymer the scanners measure: the transparent lacquered top layer or the opaque layer below, which would have a significant effect on precision measurements of carbon-fibre components.

Figure 5 also shows that System C was unable to measure the titanium coupon, which has a highly specular surface finish, for all angles. The inability of fringe projectors to measure highly specular surfaces should be of no surprise as light projected onto such reflective surfaces is insufficiently reflected into the system's cameras for measurement. It is common practice when using fringe projectors to coat specular surfaces with a diffuse coating [11], such as titanium dioxide powder. This is, however, undesirable on the production line, due to time costs, and to avoid contamination in dust sensitive aerospace manufacture environments.

SURFACE ROUGHNESS MEASUREMENTS

Scanner sensitivity to surface roughness was tested using coupons containing different sandpaper grades. A trend shown in Figure 6 may suggest increasing point cloud density with decreasing surface roughness.

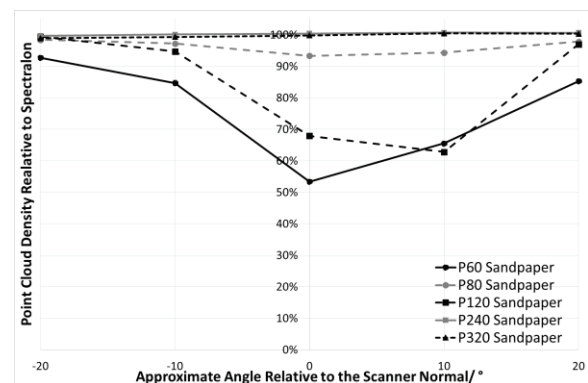


FIGURE 6 Normalised point-cloud densities for samples of different surface roughness from System C.

The point cloud densities for smoother sandpapers, with ISO grit designations [12] P320, P240 and to some extent P80, are approximately equal to their respective Spectralon reference for all measured angles, meaning that System C had no difficulty in measuring them. For the roughest sandpaper, P60, the point cloud density is

approximately half that of the reference when normal to the scanner and increases with angle. This can be explained by non-Lambertian behaviour of the larger grits such that when normal to the scanner, projected light is not sufficiently reflected into the system's cameras for measurement. Sandpapers with smoother grits may have more Lambertian behaviour and projected light is reflected such that it can be measured. The results for P120 do not fit the expected pattern and it would be reasonable to expect the results for P80 and P120 to swap places. However, as these results were highly repeatable, more work is required to investigate whether this result is unique to System C or a property of the sandpaper.

COLOUR SENSITIVITY MEASUREMENTS

Lemeš *et al* [13] have previously used RAL standard colour samples to test the effect of colour on laser scanner measurements. The National FreeForm Centre has therefore adopted a similar approach to assess the sensitivity of fringe projectors to colour.

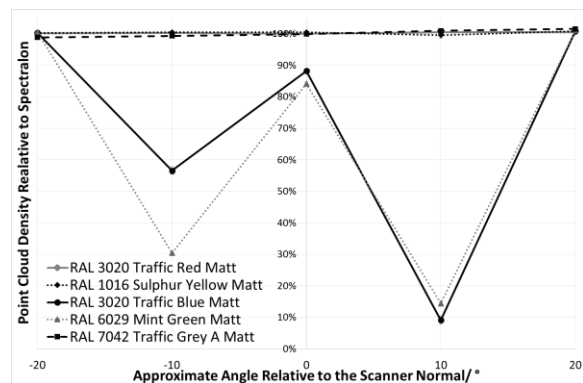


FIGURE 7 Normalised point-cloud densities for RAL standard colour samples from System C.

Figure 7 presents results from System C for sample coupons that contain matt RAL standard colour samples and shows a certain degree of colour sensitivity for this scanner. The scanner had no difficulty measuring coupons containing RAL standards: RAL 3020 Traffic Red Matt, RAL 1016 Sulphur Yellow Matt and RAL 7042 Traffic Grey A Matt; the results showing that the recorded point cloud densities were approximately equal to those of the reference coupons. The system was more sensitive to RAL 3020 Traffic Blue Matt and RAL 6029 Mint Green Matt, measuring fewer points (than the other samples) when normal to the scanner and showing marked sensitivity at the $\pm 10^\circ$ angles; interestingly the measurements appear insensitive to $\pm 20^\circ$ angles.

With the development of blue light fringe projecting optical scanners, it would be reasonable to expect these results to be reversed: greater sensitivity to the blue coloured sample coupons, which would reflect the projected blue light, and decreased sensitivity to the red coloured sample coupons, which would absorb the projected blue light. The National FreeForm Centre anticipates colour sensitivity characterisation of blue light fringe projecting optical scanners in the near future.

CONCLUSIONS

NPL's National FreeForm Centre has developed a 3D optical scanner dimensional characterisation facility to provide global industries with greater confidence in 3D optical scanners, which are rapidly replacing traditional Cartesian CMMs as they offer point cloud data capture, portability and speed. The characterisation facility contains test artefacts, test procedures and equipment to examine the performance of 3D optical surface-form scanners. This paper focuses on one of the tests to quantify 3D optical scanner's ability to measure different surface finishes (reflectances, roughnesses and colour) common to precision engineering.

A multi-faceted test artefact, the NPL 3D material coupon plate, has been developed specifically for this test to allow measurement of the test materials against highly Lambertian reference coupons through a series of fixed angles: normal to, and at $\pm 10^\circ$ and $\pm 20^\circ$ angles relative to the scanner. Results from a white light fringe projecting 3D optical scanner suggest difficulty measuring certain aluminium finishes when normal to the scanner. The same scanner was able to measurement more specular surfaces common to the aerospace industry, such as carbon fibre reinforced polymer, when normal to the sample coupon, but measured no data at other angles. These issues may be solved by application of a coating of titanium dioxide to produce a reflective diffuse surface, which is undesirable on the time-limited production line or dust sensitive aerospace manufacture environments.

Surface roughness measurements using samples with different sandpaper grits appear to show a trend of increasing point cloud density with decreasing surface roughness. The result may be explained by non-Lambertian behaviour of the sample surface with increasing grit size. However,

the results do not completely fit this suggestion and more work is required to understand whether the result is unique to this scanner model, by comparison with similar scanners, or the sample surfaces, the bidirectional reflectance distribution function (BRDF) of which could be measured.

Measurements of different coloured sample coupons show sensitivity of the scanner under test to blue and green coupons, particularly at angles just away from normal to the scanner. This suggests that objects of such colours will be more difficult to measure using this scanner, while maximum data could be expected from red and yellow coloured objects. With the development of blue light fringe projecting optical scanners, it would be reasonable to expect these results to be reversed.

NPL's 3D optical scanner dimensional characterisation facility has been developed to support optical scanner users and scanner manufacturers. As the technology develops further and more scanners become available, NPL's facility will help optical scanner users to identify suitable instruments for their needs, while for scanner manufacturers the facility provides a means of independent verification to support and demonstrate their instrument capabilities.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] Flack D, 2011. *CMM verification*. (National Physical Laboratory) Good Practice Guide No.42 <http://www.npl.co.uk/publications/cmm-verification>.
- [2] ISO 10360-2:2009 *Geometrical product specifications (GPS) -- Acceptance and reverification tests for coordinate measuring machines (CMM) -- Part 2: CMMs used for measuring linear dimensions*
- [3] COORD3. *Coordinate Measuring Machine History – Fifty Years of CMM History leading up to a Measuring Revolution*. <http://www.coord3-cmm.com/50-years-of-coordinate-measuring-machine-industry-developments-and-history/> (accessed 3 December 2014).
- [4] Ramu P, Yague J A, Hocken R J, Miller J. 2011 *Prec. Eng.* **35** 431-9.
- [5] Härtig F, Trapet E, Wäldele F, Wiegand U Traceability of Coordinate Measurements according to the Virtual CMM Concept *Proceedings of the 5th IMEKO TC-14 Symposium on Dimensional Metrology in Production and Quality Control (Zaragoza)* pp. 245-54
- [6] Acko B, McCarthy M, Haertig F, Buchmeister B. 2012 *Meas. Sci. Technol.* **23**.
- [7] Dury M, Brown S, McCarthy M, Woodward S 3D Optical Scanner Dimensional Verification Facility at the NPL's "National FreeForm Centre" *Laser Metrology and Machine Performance XI (Huddersfield)* pp. 191-200
- [8] Levoy M, et al. The digital Michelangelo project: 3D scanning of large statues *Siggraph 2000 (New Orleans, USA, 23 - 28 July 2000) (SIGGRAPH 2000 Conference Proceedings)* (New York: Assoc Computing Machinery) pp. 131-44
- [9] Clark J, Robson S. 2004 *Surv. Rev.* **37** 626-38.
- [10] RAL. *RAL Colours | RAL K5*. <http://www.ral-farben.de/en/PRODUCTS-SHOP/RAL-CLASSIC/RAL-K5.html> (accessed 4th September 2015).
- [11] Palousek D, Omasta M, Koutny D, Bednar J, Koutecky T, Dokoupil F. 2015 *Opt. Mater.* **40** 1-9.
- [12] FEPA. *FEPA - Abrasives products - Grains - P-grit sizes coated*. <http://www.fepa-abrasives.org/Abrasivesproducts/Grains/Pgritsizecoated.aspx> (accessed 4th September 2015).
- [13] Lemeš S, Zaimović-Uzunović N. Study of ambient light influence on laser 3D scanning. *7th International Conference on Industrial Tools and Material Processing Technologies*; 4 - 7 October 2009; Ljubljana, Slovenia. pp. 327-30

INFLUENCE OF THE CUTTING FORCES ON THE STRUCTURE OF A MACHINE TOOL CN -EMCO PC MILL 155-

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INSTRUCTIONS

Accuracy can be defined as the degree of agreement or conformity of a finished piece with the dimensional and geometric accuracy required. The error on the other hand, can be understood as a deviation of the edge of the tool of cutting of the theoretical position required to produce a part with the specified tolerance. The size of the error in a machine gives a measure of accuracy; it is the maximum error of translation between two points in the workload of the machine. Of course, this depends on the resolution of the system. Errors can be classified into two categories, namely the quasi-static and dynamic errors:

- Quasi-static errors are those between the tools and the workpiece which are slowly variable over time and related to the structure of the machine itself. These sources include cinematic geometric errors, which are due to own weight of the components of the machine and the thermal stresses in the structure of the machine tool.
- Dynamic errors, on the other hand are caused by sources such as the movement of the

spindle, the vibrations of the machine structure, errors due to the control of the machine etc. they are more dependent on the particular operating conditions of the machine. Quasi-static errors account for about 70% of the total error of the machine tool which makes the compensation of these errors a very important axis of research. Once the individual errors of components have been identified, the next step of the problem is to use the concept of error budget and to determine the optimum values of these errors to minimize a cost criterion [1].

Figure 1 shows a possible classification of all the factors that may affect the accuracy of machine tools. In general, the accuracy is severely limited by the geometrical structure of the machine and by the modification of this structure under static, thermal and dynamic conditions [2.3.4]. The Sources of error during machining are grouped in three categories:

- Sources of error due to the machine;
- Sources of errors due to the cutting process;
- Sources of error due to the environment.

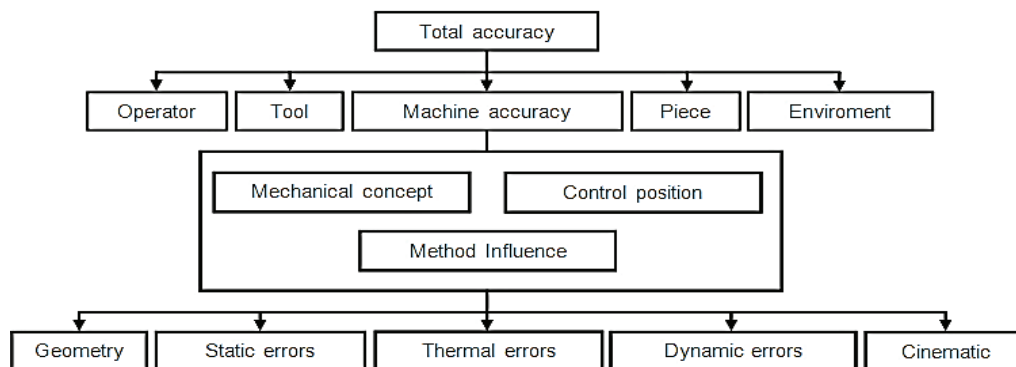


FIGURE 1. Factors affecting accuracy.

Metal cutting is a dynamic process in which the removal function solicitations have effects on the variation of quasi-static errors and can give rise to other types of errors [5]. The influence on the process accuracy can be characterized by the coexistence of the three following phenomena:

- Deformations under the effect of the cutting forces: During a machining operation, the cutting forces cause the forces on the tool and workpiece. These solicitations are propagated in the other components of the machine therefore they cause distortions that affect the dimensional and geometric quality of the finished part. These errors are a result of the variation in cutting force due to defects in shape of the brute piece and of the variation of the stiffness of the delivery system in position or structure and variation of the cutting conditions (cutting depth, cutting speed and feed rate)
- Generated temperature by the cutting: The heat generated by the cutting in the area contacting the tool / workpiece is released outside of this area. It is propagated in the structure through the mechanical contacts or cutting fluid. This heat causes distortions that degrade the overall accuracy of the machine tool.
- Vibration: The vibration is produced primarily by the cutting process. The machining accuracy obtained with the machine tools is often compromised by vibration. The relative movement between the tool and the workpiece results from the superposition of the movements controlled by the cutting movement -and movement generated by the dynamics of the process [6]. The change in this relative movement gives rise to errors that affect primarily the surface of the finished part. The variation of the depth of cut causes a variation in cutting force; which generates an excitation of the structure of the machine tool and the workpiece itself. This forced excitation of the structure generates a deflection of the cutting tool. The modeling of the cutting process becomes necessary to predict the effects of vibration for specific cutting tools [7].

CALCUTATION OF CUTTING PARAMETRES

The cutting parameters are primary variables during machining. These parameters determine the physical phenomena encountered during cutting. Their selection is directly related to the integrity of the tool and the geometric quality of the machined surface while ensuring optimum production cost.

The cutting speed (Figure 2) denoted V_c is directly related to the cutting movement. This speed is a fundamental parameter for chip formation. It depends on the machining configuration, the method and tool couple / material. Setting the cutting speed is very important because it determines the rotational frequency ω of the work piece.

$$V_c = R * \omega \quad (1)$$

The rotational frequency N of the tool (in tr.mn^{-1}) can then be deduced with the following equation:

$$N = \frac{1000 * V_c}{\pi * D} \quad (2)$$

Where R and D (mm) are respectively the radius and the final diameter of the workpiece or the tool.

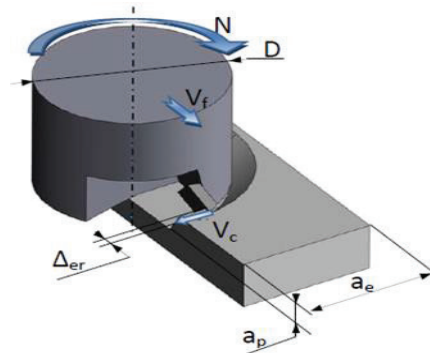


FIGURE 2. Milling operation.

The radial instant advance Δ_{er} (Figure 1) varies from 0 to the feed per tooth f . This parameter corresponds to the difference in radial displacement of the tool between the passage for the same angular position of two successive cutting edges. The feed rate is the speed of movement of the tool. This speed depends on the feed per tooth f , the number of teeth Z and the rotational frequency N .

$$V_f = f * Z * N \quad (3)$$

The cutting depth a_p is engaged in the axial depth of matter (Figure 2). The depth of the cut associated in advance, influences chip formation because it changes the chip section and therefore the energy required to shear the material. The noted lateral engagement a_e corresponds to the depth of lateral section.

To determine the forces acting on the tool we must take into account the cutting conditions and the mechanical properties of the machined material (Figure 3). During chip formation, the cutting force can be broken down into three components according to the preferred directions:

- Tangential component or main component of F_C cut: it is the component acting in the direction of the cutting speed.
- Advance or axial component F_A : the component acting in the direction of the speed of advance.
- Discharge component or radial component F_R : this is the component acting in a direction perpendicular to the other two and is acting in the direction of the axis of the cutter.

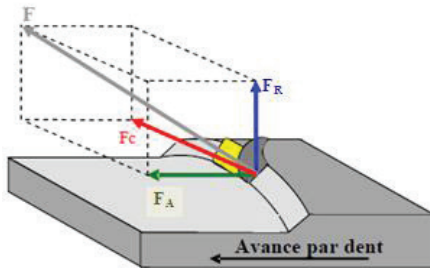


FIGURE 3. Components of milling cutting force.

The F_C cutting load depends on the specific breaking strength K_C by compression of the material being worked and its machinability, chip sizes of the tool used and the mode of work [8.9]. For the convenience of calculations, we assume the following relation to the cutting force:

$$F_C = S * K_C \quad (4)$$

With:

K_C : Specific breaking compression resistance.

S : Section of the chip defined by the advance f and depth of cut has.

And :

$$S = a * f \quad (5)$$

For advancing and penetration efforts they are given by the following equations:

$$F_A \approx \left(\frac{1}{2} \text{ à } \frac{2}{3}\right) * F_C \quad (6)$$

$$F_R \approx 0.3 * F_C \quad (7)$$

SIMULATION OF THE MECHANICAL BEHAVIOR OF THE STRUCTURE OF THE MACHINE TOOL CN

SOLIDWORKS 2012 is software which is capable of performing the various form of our CNC machine tool structure, and it allows us to register the design under various formats [10.11]. This software is a solid modeling design tool set based on functions, associative:

- Geometric Model, the most comprehensive used in CAD systems, using the information that link the geometry of the model together (example: such a surface meets with such an edge).
- Parametric, dimensions and relationships used to create a feature are captured and stored in the model; which can change quickly and without difficulty, (Change of the dimension 76 in 116 for example).

Description of the machine EMCO PC Mill 155

Vertical milling machines console type "EMCO PC Mill 155" Figure 4 are high precision machines. These milling machines are intended for milling a wide variety of pieces of steel, cast iron and non-ferrous metals, they allow the machining of the vertical and horizontal planes, grooves, angles, gear cutting, the milling wheels toothed, reamers, contours of the cams and other parts whose machining requires pivoting about the axis of the strawberries, carried out using a dividing head or a removable circular plate. The efficiency of the milling is up to the machining of parts by milling method of fast thanks to the increased speed (up to 5000 rpm). [12.13]



FIGURE 4. The vertical milling machine type - EMCO PC Mill 155-.

Conception of the machine tool CN

The mill studied in this article is newly designed to produce varied pieces. The physical prototype of the machine tool was developed by a CAO model. To establish a finite element model, the simplification of the model is necessary.

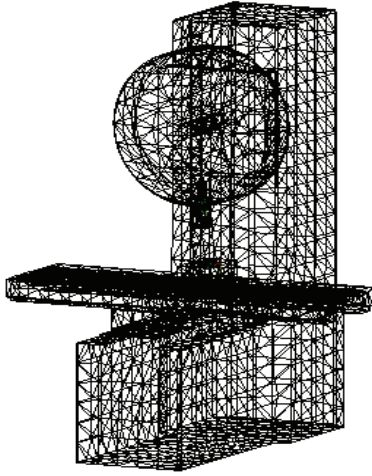


FIGURE 5. The finite element model of the machine -EMCO PC Mill 155-.

Components with less effect on the dynamic characteristics of the entire machine are deleted. Small component characteristics are also deleted. The simplified model is presented in Figure 5. According to the practical constraints, the lower part of the machine tool is fully restricted. The relative position between the contact surfaces is constant in a certain configuration. Therefore, the contact surfaces are defined as bond contacts. The rigidity of contact is not updated during the calculation process. The initial rigidity can be automatically obtained on the platform SOLIDWORKS by the contact factor, the size of the element and the elasticity of the contact materials module.

The stiffness will be adjusted in the update process of the FE model according to the results of the simulation of the machine tool. We first start by designing the structure of the machine then the structural simulation to determine the strength and the rigidity by establishing an account of constraints and component deformations. The type of structural analysis that we do depends on the operation machining and plant material. We used the data summarized in the following table:

TABLE 1. Cutting parameters

Material to be machined	Depth of cut [mm]	Cutting forces [N]	H_m [mm]					
			0.1	0.2	0.4	0.6	0.8	1
Not allied steel XC 35 $H_B=500\text{à }600 \text{ N/mm}^2$	0.5	K_C	3150	2640	2210	1970	1850	1750
		F_C	146,48	122,76	102,77	91,61	86,03	81,38
		F_A	97,65	81,84	68,51	61,07	57,35	54,25
		F_R	43,94	36,83	30,83	27,48	25,81	24,41
	1	K_C	3150	2640	2210	1970	1850	1750
		F_C	292,95	245,52	205,53	183,2	172,05	162,75
		F_A	195,30	163,68	137,02	122,1	114,70	108,50
		F_R	87,89	73,66	61,66	54,96	51,62	48,83

RESULTS

Figure 6 shows the displacements of the frame of the machine tool due to the cutting forces for a material not allied steel XC 35 and thickness of the chip $H_m = 0.1\text{mm}$.

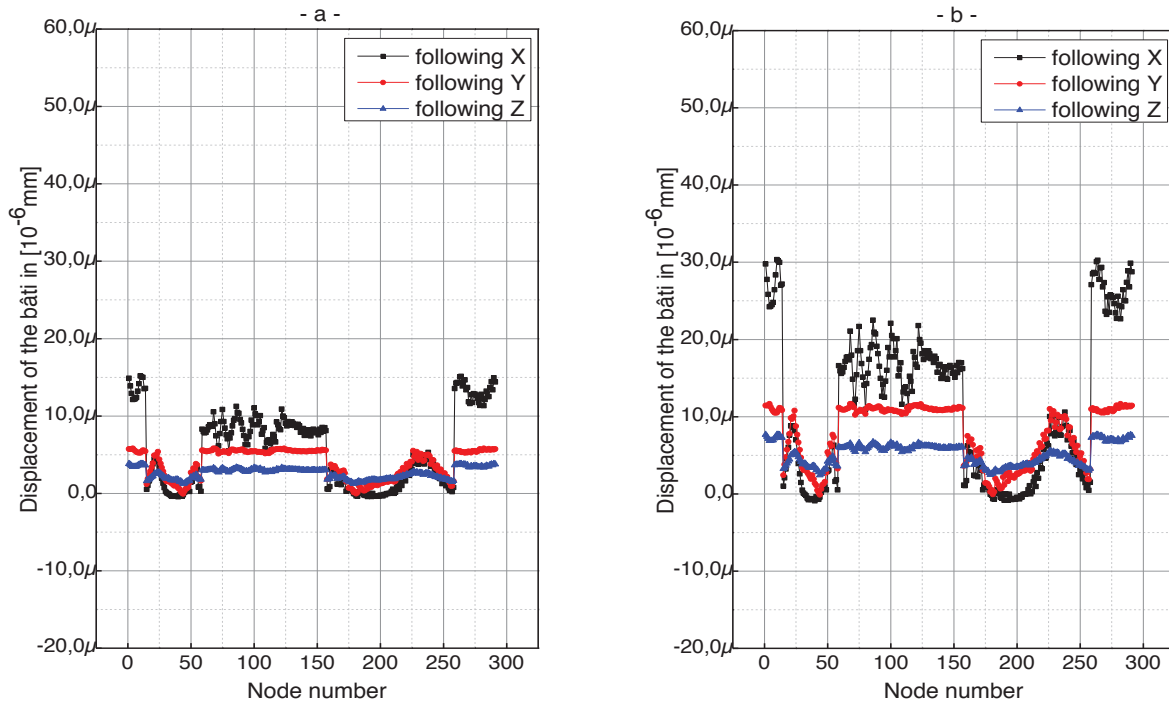


FIGURE 6. Variation of displacement of the frame relative to the infeed depth, a- $f = 0.5\text{ mm}$, b- $f = 1\text{ mm}$.

Figure 7 shows the displacements of the carriage of the machine tool due to the cutting forces for a material not allied steel XC 35 and thickness of the chip $H_m = 0.1\text{ mm}$.

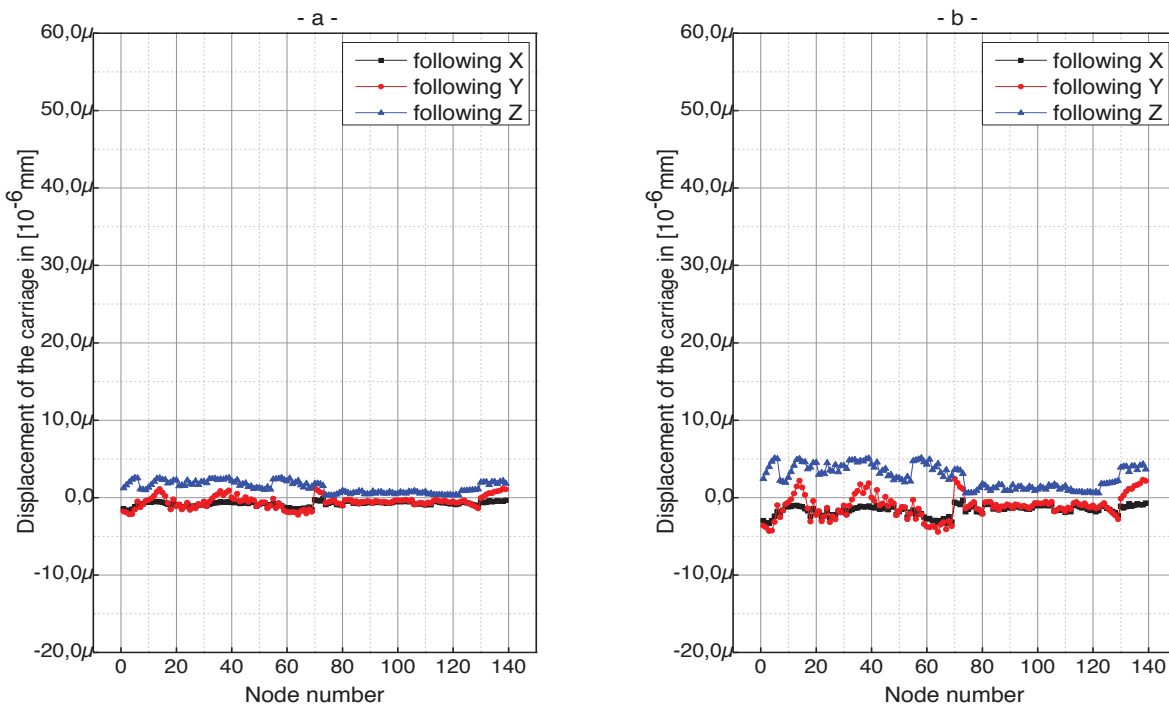


FIGURE 7. Variation of displacement of the carriage relative to the infeed depth, a- $f = 0.5\text{ mm}$, b- $f = 1\text{ mm}$.

Figure 8 graphically illustrates the displacement of the table due to cutting forces for a material not allied steel XC 35 and thickness of the chip $H_m = 0.1\text{mm}$.

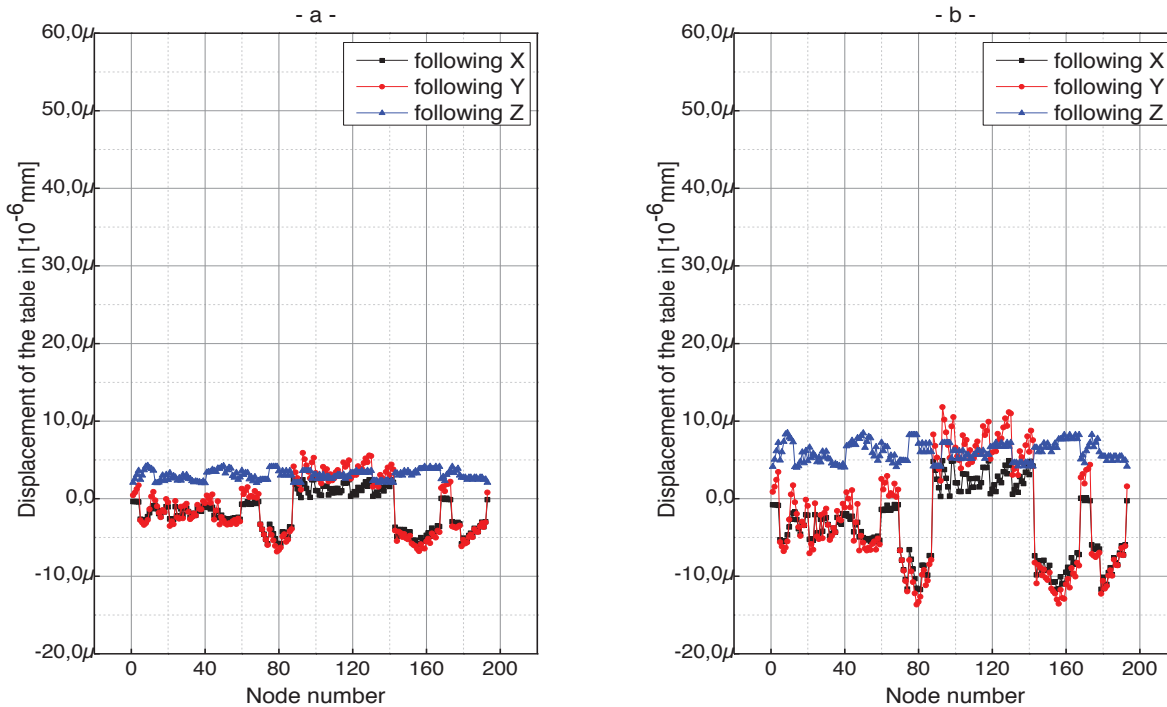


FIGURE 8. Variation of displacement of the table relative to the infeed depth, a-f = 0.5mm, b- f =1 mm.

DISCUSSTION OF THE RESULTS

From Figure 6, we see that the displacement of the frame in the three directions when the depth augments password augment, and we also note that the displacement along the X axis is larger than the other Y and Z axes by that the force of the cutting tool and in the direction of the axis X. It is noted that the given displacement gap Y relative to the deviations of the displacement X and Z augments when the depth of cut augment. See table 2

An analysis of Figure 7, it can be deduced that when plunging depth augment, there is an increase of displacement of the carriage along

the three directions. And we notice that the displacement along the Z axis is greater than the other X and Y axes because effort of penetration of the tool is in the direction of the axis Z. See table 2

Note that the movement of the figure 8 table following the three directions grows when the depth of cut grows, and we also find that the displacement along the Z axis is nearly constant. That is observed following the displacement gap Y is greater than the displacement following gap X. See table 2

TABLE 2. Variation of displacement of the structure of the machine tool as a function of depth of cut.

Structure of the machine		frame		Chariot		Table	
Depth of cut [mm]		0.5	1	0.5	1	0.5	1
Displacement in 10^{-6} mm	X	15.03	30.34	1.66	3.34	-5.85	-11.71
	Y	5.82	11.65	2.21	3.86	-6.83	-13.66
	Z	3.82	7.65	2.53	5.10	4.23	8.46

CONCLUSION

The loss of accuracy in machine tools with three axes is due to geometric imperfections of the mechanical structure and the changes under the influence of static, thermal and dynamics conditions. In this article, we presented the design of the structure of the machine tool CN by SOLIDWORKS software. Next, a structural simulation by varying the cutting parameters to determine the strength and stiffness by establishing an account of constraints and component movements. The type of structural analysis we do depends on the machining operation and milled material. Finally, the results showed that the displacements and deformations of the machine tool structure increases linearly with increasing depth of cut, and decrease with increasing the thickness of the chip.

REFERENCES

- [1] B. Kuhlenkötter, M. Sdahl, Automated inspection system for headlamp reflectors ,International Journal of Advanced Manufacturing Technology 32 (500–504), Springer, 2007.
- [2] K. Tong, E. A. Lehtihet, S. Joshi, Software compensation of rapid prototyping machines ,Precision Engineering 28 (280–292), Elsevier, 2004.
- [3] J. K. Rai, P. Xirouchakis, Finite element method based machining simulation environment for analyzing part errors induced during milling of thin-walled components, International Journal of Machine Tools and Manufacture 48 (629–643), Elsevier, 2008.
- [4] A. Dugas, J. J. Lee, and J. Y. Hascogt, An Enhanced Machining Simulator with Tool deflection Error Analysis, Journal of Manufacturing Systems 21/ No. 6, 2002.
- [5] R. J. Hocken, Automated Production Technology, Annals of CIRP, 1990.
- [6] V. Ragunath, Thermal effects on the accuracy of numerically machine tools, Purdue University, Indiana, 1985.
- [7] A.H. Slocum, Precision machine design, Prentice-Hall Englewood Cliffs, New Jersey, 1992.
- [8] Gaëtan. A, Identification and modeling of torsor milling cutting actions, Université Bordeaux 1, 2010.
- [9] M. Richard, cutting forces - cutting power and absorbed power, France, 2007.
- [10] Getting to Know SolidWorks PDF (Academy of Orleans-Tours)
- [11] Tutorial SolidWorks 2012
- [12] EMCO MAIER, Edition A2001-04, "EMCO PC Mill 155: milling machine controlled by PC for training," Austria, Ref.-No. EN 4345.
- [13] S. Merghache, A. Ghernaout, Qualification and validation the metrology module located on CN machine tool -EMCO PC Mill 155 –, Journal of Physics: Conference Series 483(1–12), IOP science 2014

PERFORMANCE EVALUATION OF A LASER TRACKER HAND HELD TOUCH PROBE

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INTRODUCTION

Hand held touch probes and laser trackers are increasing the scope and applicability of laser trackers. While methods to evaluate the performance of laser trackers in conjunction with spherically mounted retroreflectors (SMRs) are well established, methods for evaluating the performance of hand held probes [1-2] are still under discussion within ISO 10360-10. We discuss the performance of a hand held touch probe in this paper. Single point articulation tests (SPATs) are commonly employed both to calibrate and to test these devices. We have modeled a hand held touch probe and subsequently performed simulations to understand the influence of different error parameters on measured coordinates in two different configurations of SPATs. The overall objective is to develop detailed uncertainty budgets for measurements made using the hand held touch probe. We present preliminary simulation and experimental results here as a first step towards realizing that objective.

THE HAND HELD TOUCH PROBE

There are several designs of hand held touch probes in the market. A schematic of the hand held touch probe under evaluation is shown in Fig. 1. The laser tracker measures the position of the retroreflector located in the hand held touch probe. An orifice at the apex o of the retroreflector allows a portion of the laser beam to travel further onto a charge-coupled device (CCD). The position of the laser spot on the CCD determines the pitch angle β (rotation about the y axis of the hand held touch probe) and yaw angle γ (rotation about the z axis). Roll angle α (rotation about the x axis) is determined by a gravity sensor. The link lengths a (along the x axis), b (along the y axis, not shown in the figure because it is nominally zero in this configuration), and c (along the z axis) are determined through a calibration procedure performed prior to measurement. Using the three measured angles and three link lengths, the coordinates of the stylus tip P can be

determined through a geometric transformation. The hand held touch probe also has a separate yaw joint. While we use this yaw joint to orient the retroreflector towards the tracker between the different SPATs, we have not exercised the yaw joint during a SPAT; we therefore do not consider this in our model. Simulations and experiments were performed using the horizontal stylus only.

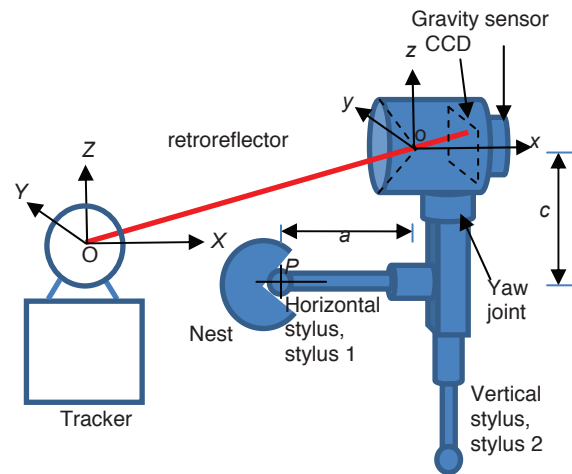


FIGURE 1. Schematic of the hand held touch probe

COORDINATE SYSTEMS

We define two coordinate systems, one on the tracker (XYZ) and another on the hand held touch probe (xyz). The origin of the coordinate system on the hand held touch probe is located at the apex of the retroreflector. The x axis is normal to the plane of the CCD while the z axis is parallel to the long handle of the hand held touch probe as shown in Fig. 1. Let the position of the retroreflector as recorded by the tracker in spherical coordinates be (R, H, V) . In order to determine the coordinates of the stylus tip P in the laser tracker coordinate system, we employ the following sequence of translations and rotations.

Let the retroreflector of the hand held touch probe first be located at $(R, 0, 0)$ as shown in Fig. 2(a). Let the orientation of the hand held touch probe be such that roll, pitch, and yaw angles are zero at this position. The stylus tip coordinate is known in the tracker frame at this position and is given by $(R-a, -b, -c)$. The hand held touch probe is then rotated by an angle V about the Y axis and then by an angle H about the Z axis to the position shown in Fig. 2(b). The hand held touch probe is then rotated about its x axis by the roll angle (Fig. 2(c)) and subsequently about its y axis by the pitch angle (Fig. 2(d)) and then about its z axis by the yaw angle (not shown in Fig. 2). The resulting coordinate for the stylus tip is the desired coordinate in the laser tracker coordinate system.

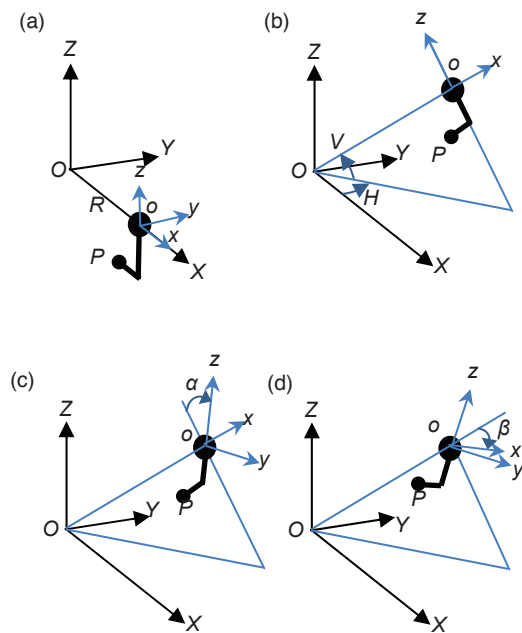


FIGURE 2. Transformations to calculate stylus tip coordinate in laser tracker frame (a) retroreflector located at $(R, 0, 0)$ with hand held touch probe oriented such that there is no roll, pitch, or yaw, (b) hand held touch probe rotated about Y axis by angle V and then about the Z axis by angle H , (c) hand held touch probe rotated about its x axis by the roll angle, (d) hand held touch probe rotated about its y axis by the pitch angle

Single point articulation tests

Two configurations of SPATs are considered in this study; they are shown in Fig. 3. In each

case, with the stylus tip located in the nest so that the center of the stylus tip remains in the same position during articulation, the hand held touch probe is rotated about each of the three axes to the extent possible, which is $\pm 30^\circ$ for the pitch and yaw axes and $\pm 60^\circ$ for the roll axis. While the hand held touch probe itself is capable of 360° along the roll axis, physical limitations in the test setup only allowed for $\pm 60^\circ$ in that axis. The nominal values for link lengths a , b , and c are 85 mm, 0, and 85 mm respectively for SPAT #1, similar to the horizontal configuration shown in Fig. 1 and in Fig. 3(a), and 85 mm, 40 mm, and 85 mm respectively for SPAT #2 as shown in Fig. 3(b). The nest was located about 2 m from the laser tracker.

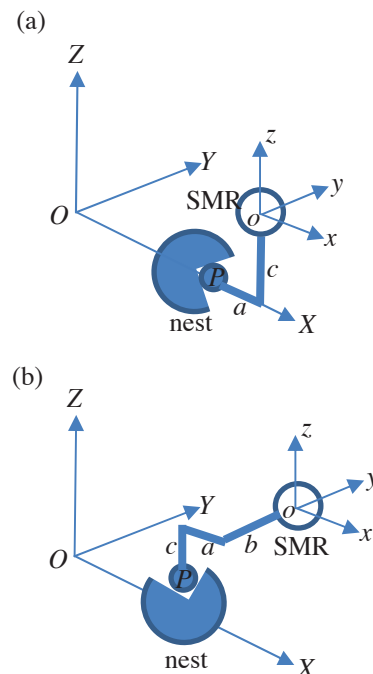


FIGURE 3. Two configurations of SPATs shown (a) SPAT #1 ($a = 85$ mm, $b = 0$, $c = 85$ mm) and (b) SPAT #2 ($a = 85$ mm, $b = 40$ mm, $c = 85$ mm)

Errors in the link lengths and in the measured angles produce errors in the measured coordinates. In order to determine reasonable values for the parameters to be used as input to the simulations, two different experiments were performed. First, the manufacturer suggested calibration procedure was performed several times to determine the one standard deviation repeatability in the link lengths. This procedure involves performing a SPAT while exercising the

yaw joint between the different orientations of the hand held touch probe. The manufacturer's software considers the different stylus tip coordinates obtained during the SPAT and the laser tracker's determination of the nest coordinate obtained using an SMR to evaluate the link lengths. Multiple such calibrations resulted in link length repeatability on the order of 10 μm . Although it is possible there are other sources of systematic error that will result in a larger uncertainty in the link lengths, the one standard deviation repeatability of 10 μm is considered as the standard uncertainty in each of the three link lengths a , b , and c , and propagated to the stylus tip in the simulations.

In order to estimate the uncertainty in the roll, pitch, and yaw angles, the hand held touch probe was mounted on a precision air bearing rotary table and the yaw angle of the hand held touch probe was compared against the encoder readings of the rotary table. That experiment indicated a yaw angle error on the order of 1 mrad, which is considered as the uncertainty in each of the three angles, and propagated to the stylus tip in the simulations. It should be noted that the roll angle is a coarser measurement in comparison to pitch and yaw; at this time, we have not quantified the errors associated with

the roll angle and therefore simply use the same value as obtained for yaw.

Results and discussion

As mentioned in the previous section, the purpose of the simulations is to understand the influence of errors in each of the six input parameters on the stylus tip coordinate for the two different SPATs. A SPAT can be performed in either the absolute mode or in the relative mode. In the absolute mode, the coordinates of the stylus tip for the various orientations in a SPAT are compared to the coordinate as measured using an SMR mounted on the nest. In the relative mode, the coordinates of the stylus tip for the various orientations in a SPAT are compared against each other.

The stylus tip size in our hand held touch probe was 6 mm in diameter. The SPAT is typically performed using a manufacturer provided nest that has a conical seat in a 1.5 in spherical shell. When the stylus tip is located in the conical seat, the center of the tip is ideally also the center of the spherical shell, and the coordinate of that point can be determined using a 1.5 in SMR. In some situations, it may not be physically possible to measure the coordinate of the nest using an SMR and therefore relative SPATs are sometimes performed.

TABLE 1. Simulation results for SPAT #1 analyzed in absolute mode and in relative mode. All units in millimeters.

	SPAT #1 in absolute mode									SPAT #1 in relative mode								
	Roll test			Pitch test			Yaw test			Roll test			Pitch test			Yaw test		
	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z
a	0.01	0.00	0.00	0.01	0.00	0.01	0.01	0.01	0.00	0.00	0.00	0.00	0.00	0.00	0.01	0.00	0.01	0.00
b	0.00	0.01	0.01	0.00	0.01	0.00	0.01	0.01	0.00	0.00	0.00	0.01	0.00	0.00	0.00	0.01	0.00	0.00
c	0.00	0.01	0.01	0.01	0.00	0.01	0.00	0.00	0.01	0.00	0.01	0.00	0.01	0.00	0.00	0.00	0.00	0.00
α	0.00	0.08	0.07	0.00	0.11	0.00	0.00	0.08	0.04	0.00	0.02	0.07	0.00	0.05	0.00	0.00	0.00	0.04
β	0.08	0.07	0.09	0.12	0.00	0.12	0.08	0.00	0.09	0.00	0.07	0.03	0.05	0.00	0.05	0.00	0.00	0.01
γ	0.00	0.08	0.07	0.00	0.08	0.00	0.04	0.08	0.00	0.00	0.03	0.07	0.00	0.00	0.00	0.04	0.01	0.00

TABLE 2. Simulation results for SPAT #2 analyzed in absolute mode and in relative mode. All units in millimeters.

	SPAT #2 in absolute mode									SPAT #2 in relative mode								
	Roll test			Pitch test			Yaw test			Roll test			Pitch test			Yaw test		
	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z	X	Y	Z
a	0.01	0.00	0.00	0.01	0.00	0.01	0.01	0.01	0.00	0.00	0.00	0.00	0.00	0.00	0.01	0.00	0.01	0.00
b	0.00	0.01	0.01	0.00	0.01	0.00	0.01	0.01	0.00	0.00	0.00	0.01	0.00	0.00	0.00	0.01	0.00	0.00
c	0.00	0.01	0.01	0.01	0.00	0.01	0.00	0.00	0.01	0.00	0.01	0.00	0.01	0.00	0.00	0.00	0.00	0.00
α	0.00	0.10	0.10	0.00	0.12	0.04	0.00	0.09	0.08	0.00	0.06	0.09	0.00	0.05	0.00	0.00	0.00	0.05
β	0.09	0.07	0.08	0.12	0.00	0.12	0.09	0.00	0.09	0.00	0.07	0.03	0.05	0.00	0.05	0.00	0.00	0.03
γ	0.04	0.09	0.07	0.04	0.09	0.02	0.08	0.09	0.00	0.00	0.03	0.07	0.00	0.00	0.02	0.05	0.03	0.00

The results of the simulations for the two SPATs are shown in Tables 1 and 2. Each of the SPATs are comprised of three tests – a roll test, a pitch test, and a yaw test. Each of these three tests involves probing the nest by rotating the hand held touch probe along one of the three axes. The tables show the *maximum absolute error* in stylus tip along the X, Y, and Z axis for each of these three simulated tests. Each row in the table corresponds to stylus tip error in the presence of an error in one of the six input parameters (a 10 μm error in a , b , or c , or a 1 mrad error in α , β , or γ). The data are analyzed in absolute mode and again in relative mode.

Tables 1 & 2 show that unit errors in the link length reflect as unit errors in one or more of the coordinates; that is, there is no amplification of the errors as expected. Angular errors, however, depend on the Abbe offset and can produce large point coordinate errors. A pitch of 1 mrad, for example, produces a 0.12 mm error along X and Z for both SPAT configurations in absolute mode for the hand held touch probe under consideration. Those errors drop to 0.05 mm in relative mode, indicating that the relative mode of analysis may attenuate the effect of certain error sources.

There are additional interesting observations that can be made from these tables. A roll test is not necessarily the most sensitive test to detect an error in the roll angle α . In fact, the Y coordinate in a pitch test is more sensitive to error in the roll angle for the hand held touch probe under consideration. While a pitch test is sensitive to pitch angle errors, yaw angle errors can be captured in any of roll, pitch, or yaw tests. In addition, it can be seen that Tables 1 and 2 are nearly identical indicating that the Y offset of 40 mm in SPAT #2 did not produce a noticeable amplification of angular errors in comparison to SPAT #1.

Table 3 shows the experimentally obtained maximum absolute errors along the three axes for the roll, pitch, and yaw test for the two configurations of SPATs. The experiments were repeated three times; the absolute maximum errors from all three repeats are shown in the table. The data for these tests were analyzed in relative mode because the nest used to acquire data was a three-pronged seat for a 6 mm tip that could not seat an SMR. The experimentally observed errors are fairly small, with the largest error on the order of 0.06 mm. It can be seen

that the results obtained from simulations (right half of Tables 1 and 2 that show SPAT results based on relative mode of analysis) are also on the order of about 0.07 mm, indicating that the values of input parameters used in the simulations are reasonable. However, as mentioned earlier, we do note that it is possible that the relative mode of analysis has suppressed the effect of certain error sources and therefore the hand held probe may possess error sources that have not been revealed. We plan on performing these tests again in absolute mode in the future.

TABLE 3. Experimentally obtained SPAT errors in millimeters. Data analyzed in relative mode.

SPAT #1			
	X	Y	Z
Roll test	0.03	0.05	0.02
Pitch test	0.04	0.03	0.03
Yaw test	0.04	0.06	0.06
SPAT #2			
	X	Y	Z
Roll test	0.03	0.03	0.02
Pitch test	0.04	0.03	0.02
Yaw test	0.03	0.06	0.02

CONCLUSIONS

SPATs are commonly employed to calibrate and evaluate the performance of hand held touch probe accessories of laser trackers. We describe a model based approach to understand the influence of different parameters on stylus tip errors for different configurations of SPATs. As future work, we plan on refining the model and determining more suitable values for input parameters of the simulation, such as better estimates for roll angle errors. We also plan on performing absolute SPATs and length tests along sensitive directions. Characterizing error sources is critical towards developing test procedures that are sensitive to the different error sources.

REFERENCES

- [1] R Kupiec, R Dubno, and J Sladek, Accuracy assessment of a laser tracker system, 11th ISMQC, September 11-13 2013, Cracow-Kielce, Poland
- [2] K Lau, Y Yang, Y Liu, and H Song, Dynamic performance evaluation of 6D laser tracker sensor, PerMIS'10, September 28-30, 2010, Baltimore, MD

SUPPRESSION OF TOOL WEAR IN ULTRA-PRECISION DIAMOND TURNING OF STEEL BY NITRIDING

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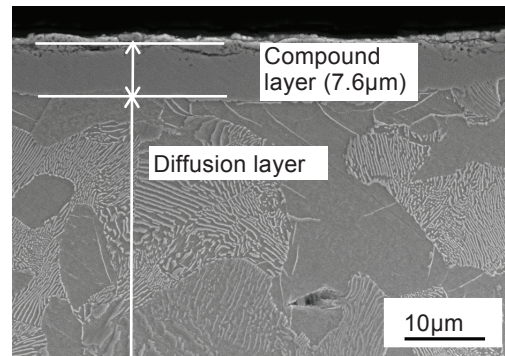
BACKGROUND

Demand has been increasing for highly durable molds of heat-resistant materials for complex and precise optical components with high aspect ratio. Diamond is an ideal cutting tool material for high efficient ultraprecision metal cutting of complex three-dimensional metal works. However, it is well known that diamond cutting tool shows severe wear in cutting of heat-resistant materials such as ferrous metals. Ab-initio calculation by the authors showed that the essential mechanism of severe wear is the dissociation of carbon atoms on diamond surface due to the chemical interaction with iron atoms in workpiece [1]. Establishing a chemical bond between the iron and another chemical element is an effective method to avoid the chemical reaction between the iron and carbon atoms. Brinksmeier et al. [2] applied the idea for industrial use by nitriding the subsurface zone of steel and showed that the diamond machining of steel with optical surface quality is possible.

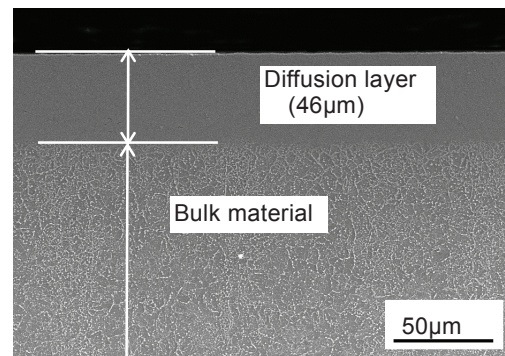
In this study, to obtain a practical nitriding conditions for the fabrication of precise molds, effect of nitriding on suppression of tool wear were investigated.

PLASMA NITRIDING OF THE WORKPIECES AND CUTTING EXPERIMENTS

The cylindrical workpieces of C45E (0.45%C steel, JIS/S45C) and X30Cr13 (13%Cr stainless steel, JIS/SUS420J2) modified were used in the cutting experiments. The latter is a material of the mold for plastic forming. The outer and inner diameters and length of



(a) C45E



(b) X30Cr13 modified

FIGURE 1. SEM images of cross section of nitrided layers.

workpieces were 36mm, 16mm, and 25mm, respectively.

The plasma nitriding was carried out on C45E at a temperature of 783K for 4 hours. Figure 1(a) shows the SEM image of cross section of nitride layer of the workpiece after polishing and etching with nital reagent. The compound layer of 7.6μm thickness on the workpiece

surface and the diffusion layer of $160\mu\text{m}$ depth beneath the compound layer were observed.

The plasma nitriding of X30Cr13 modified was carried out at a temperature of 823K for 10 hours. Figure 1(b) shows the SEM image of cross section of nitride layer of the workpiece after polishing and etching with Marble's reagent. On the surface of workpiece, thin white layer of less than $1\mu\text{m}$ thickness is observed. However, the thickness and structure of the layer were not uniform. The diffusion layer of $46\mu\text{m}$ depth was formed beneath the surface layer.

As shown in Figure 2, the face turning experiments were carried out on an ultra-precision lathe using a chevron nosed diamond cutting tool with 130° degree nose angle. The spindle speed and feed rate were 500min^{-1} and $10\mu\text{m}/\text{rev}$, respectively.

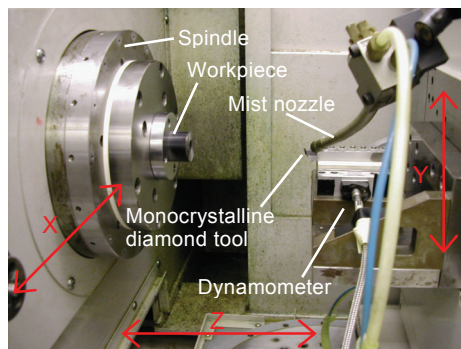


FIGURE 2. Face turning of the workpiece.

EXPERIMENTAL RESULTS OF C45E CUTTING

Figure 3 shows the appearances of C45E specimens with a depth of cut of $3\mu\text{m}$ after a cutting distance of 160m . The nitrided specimen shows smooth and glossy surface while unnitrided specimen shows rough surface like a satin surface. Figure 4 shows the SEM images of finished surfaces of compound layer of nitrided C45E comparing with that of unnitrided one after a cutting distance of 160m . Figure 5 shows the surface profiles corresponding to the surfaces shown in Figure 4 measured by a stylus type surface tester.

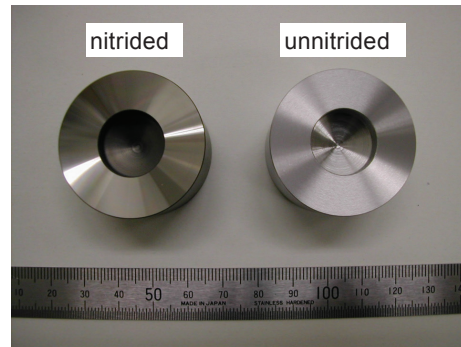
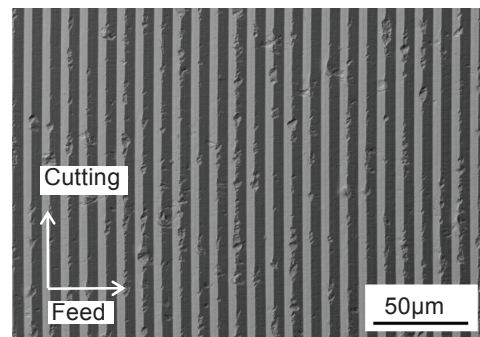
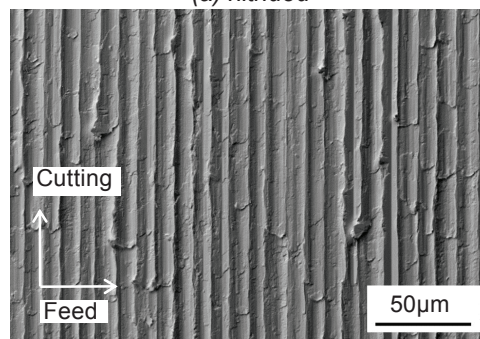


FIGURE 3. Appearances of C45E specimens after a cutting distance of 160m .



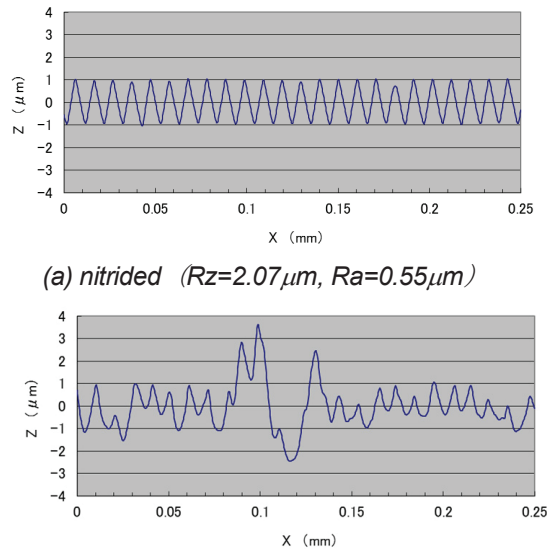
(a) nitrided



(b) unnitrided

FIGURE 4. SEM images of finished surface of C45E after a cutting distance of 160m .

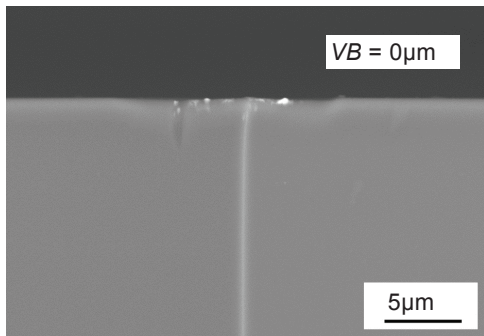
The finished surface of nitrided specimen is smooth and the feed marks are clearly observed as shown in Figure 4(a). The surface roughness is $2.07\mu\text{mRz}$ which is a little smaller than the theoretical value of $2.33\mu\text{mRz}$ as shown in Figure 5(a). The decrease of surface roughness may be the result of scraping away around the peak of feed marks by a hard nitride as shown in Figure 4(a). On the other hand, many impressions of surface tearing off and/or adhesion are observed on the finished surface



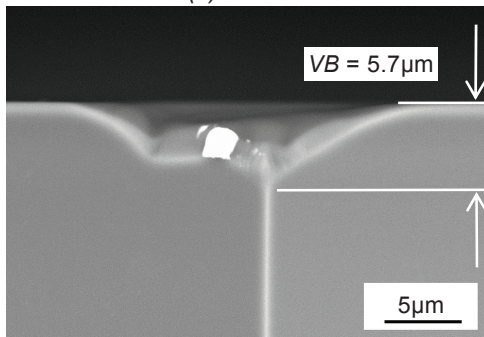
(a) nitrided ($R_z=2.07\mu\text{m}$, $R_a=0.55\mu\text{m}$)

(b) unnitrided ($R_z=6.09\mu\text{m}$, $R_a=0.72\mu\text{m}$)

FIGURE 5. Surface profiles of C45E after a cutting distance of 160m.



(a) nitrided



(b) unnitrided

FIGURE 6. Comparison of flank wear after 160m cutting of C45E.

of unnitrided specimen as shown in Figure 4(b). The surface roughness is $6.09\mu\text{m}R_z$ that is about 2.6 times as large as theoretical value as shown in Figure 5(b).

Figure 6 shows the SEM images of flank wear of cutting tools after a cutting distance of 160m. No wear can be observed on the tool which finished nitrided specimen, while large flank wear is observed on the tool which finished unnitrided one.

EXPERIMENTAL RESULTS OF X30Cr13 MODIFIED CUTTING

Figure 7 shows the appearances of X30Cr13 modified specimens with a depth of cut of $10\mu\text{m}$ after a cutting distance of 80m. In the same manner as C45E specimens, the nitrided specimen shows smooth and glossy surface while unnitrided specimen shows rough surface like a satin surface.

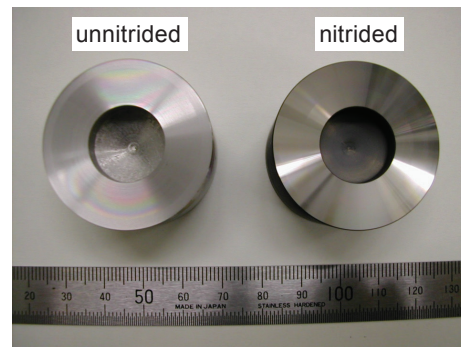
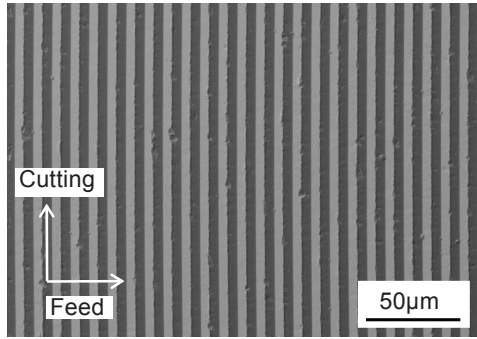


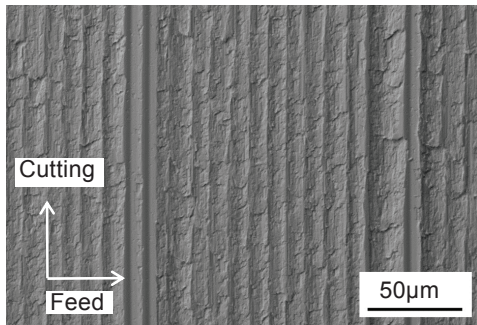
FIGURE 7. Appearances of X30Cr13 modified.

Distinction between the finished surfaces of nitrided and unnitrided specimens of X30Cr13 modified after a cutting distance of 320m are shown in Figure 8 and the surface profiles of the specimens are shown in Figure 9.

The finished surface of nitrided specimen is smooth and feed marks are clearly observed as shown in Figure 8(a). The surface roughness is $2.31\mu\text{m}R_z$ which is almost same as theoretical value of $2.33\mu\text{m}R_z$. While, rough surface with many impressions of surface tearing off and/or adhesion are observed on the unnitrided specimen as shown in Figure 8(b). The surface roughness is $3.82\mu\text{m}R_z$. In Figure 8(a), around the peak of feed marks, some impressions of scraping away are observed. However, the number of impressions is less than that in case of C45E.

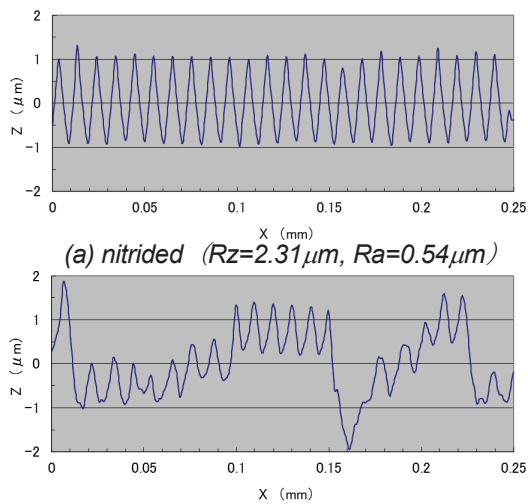


(a) nitrided



(b) unnitrided

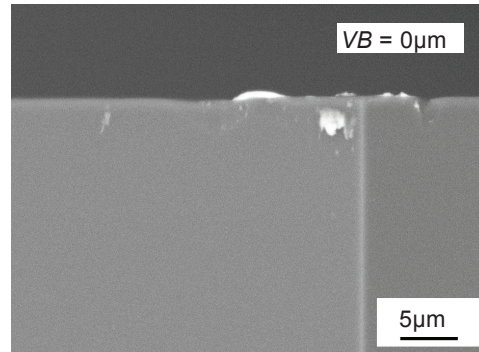
FIGURE 8. SEM images of finished surface of X30Cr13 modified (cutting distance: 320m).



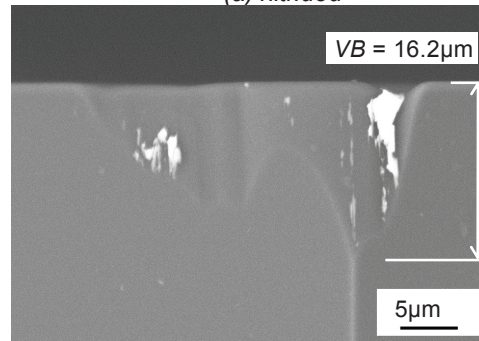
(b) unnitrided ($R_z=3.82\mu\text{m}$, $R_a=0.63\mu\text{m}$)

FIGURE 9. Surface profiles of X30Cr13 modified.

Figure 10 shows the SEM images of flank wear of cutting tool after a cutting distance of 80m. No wear can be observed on the tool which finished nitrided specimen, while large flank wear is observed on the tool which finished unnitrided specimen.



(a) nitrided



(b) unnitrided

FIGURE 10. Comparison of flank wear after 80m cutting of X30Cr13 modified.

Figure 11 and 12 show the change in width of flank wear V_B and surface roughness R_z , respectively, with cutting distance L in cutting of X30Cr13 modified. Both of V_B and R_z increase in cutting of unnitrided specimens. On the other hand, in cutting of nitride specimens, V_B is almost constant until the cutting distance reaches 320m and also the surface roughness is same as the theoretical value.

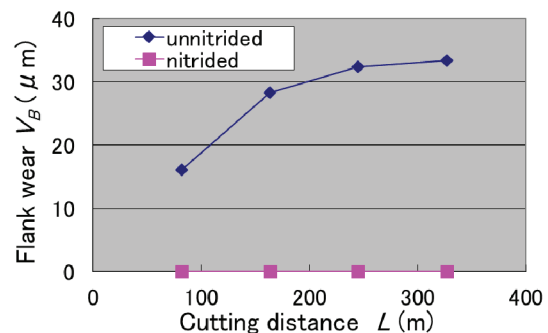


FIGURE 11. Change in the flank wear with cutting distance (X30Cr13 modified).

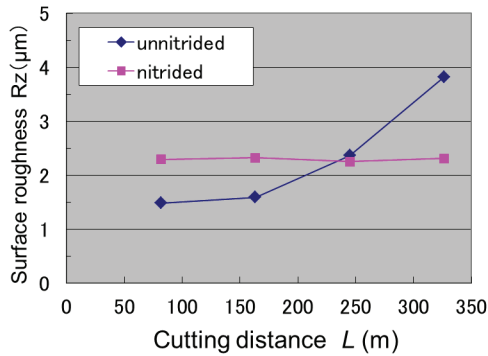


FIGURE 12. Change in the surface roughness R_z with cutting distance (X30Cr13 modified).

Figure 13 and 14 show the nitrogen density distribution estimated by energy dispersive X-ray spectroscopy (EDX) and Knoop hardness, respectively, beneath the specimen surface of nitrided X30Cr13 modified. The nitrogen density and Knoop hardness are almost constant until about $45\mu\text{m}$ depth beneath the surface. In the cutting experiment, the total depth of cut was $40\mu\text{m}$ from the original specimen surface. Therefore, in the diffusion layer with sufficient nitrogen, wear of diamond cutting tool was strongly suppressed.

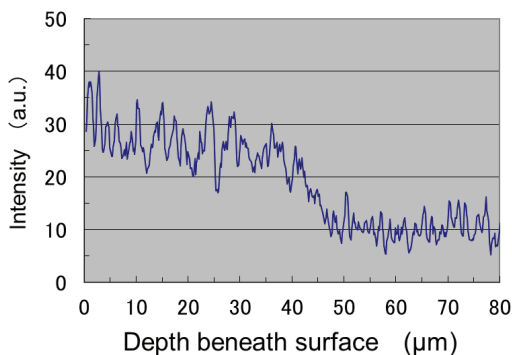


FIGURE 13. EDX intensity of nitrogen beneath surface (X30Cr13 modified).

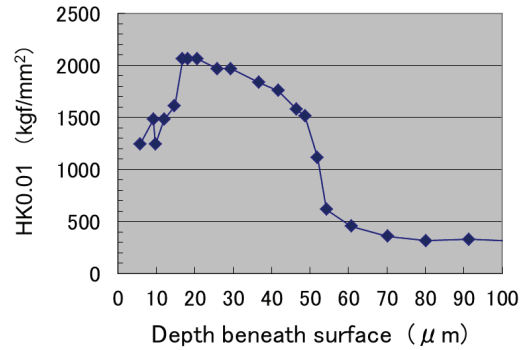


FIGURE 14. Knoop hardness beneath surface (X30Cr13 modified).

SUMMARY

In ultraprecision diamond cutting of steel, if the surface layer of workpiece is properly nitrided, the wear of diamond tool is remarkably suppressed. Using nitriding process, the highly durable molds with high heat-resistant for complex and precise optical components can be successfully finished by diamond turning.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] Shimada, S., Tanaka, H., Higuchi, M., Yamaguchi, T., Honda, S., Obata, K., "Thermo-Chemical Wear Mechanism of Diamond Tool in Machining of Ferrous Metals", 2004, Annals of the CIRP, 53, 1, 57-60.
- [2] Brinksmeier, E., Gläbe, R., Osmer, J., "Ultra-Precision Diamond Cutting of Steel Molds", 2006, Annals of the CIRP, 55, 1, 551-554.

Artifact Based Diamond Ball Mill Error Correction for Freeform Optics Manufacturing

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INTRODUCTION

In the diamond machining of optics, form tolerance requirements require a robust method for correcting the errors induced by the tool. For turning of axi-symmetric optics or freeforms with small departures from spherical, these methods are relatively well established and various on- and off-machine schemes are available. However, demand for freeform optics of general shape is growing, and machining of these optics requires freeform ball-milling techniques. This paper describes a general spherical artifact-based procedure for separating inherent ball-mill errors (waviness and decentering) that can then be used to accurately cut optics that depart in shape substantially from the spherical artifact.

One technique used for correcting a ball end mill milling tool path is to cut the desired optic, measure it, subtract the error from the tool path, and then remachine the optic. This works well when you are only planning to make a single optic with a single optical prescription. However, if the same tool is to be used to generate multiple optics with different prescriptions or if the prescription is not easily measured (freeforms), correcting each specific optic is cumbersome or impossible. Here we show a procedure that models the actual tool errors and uses them to correct the tool path before cutting the optical prescription.

In conventional ball-milling, the errors in the machine motions themselves often dominate the overall uncertainty in the geometry of the final component. However, in ultra-precision ball-milling, the machining

centers are typically more accurate than the tools. Therefore, in order to correct the dominate sources of form error on ultra-precision diamond milled parts a procedure must be developed to calibrate the errors of the tool. An artifact based procedure was developed to map and correct the tool errors before creating the final desired optical shape.

Two major error sources on a milling tool are diamond position relative to the axis of rotation (radial shift) and the shape of the cutting edge (waviness), shown in Figure 1.

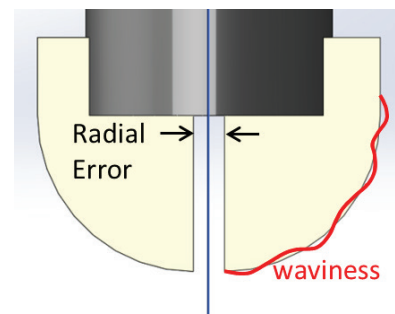


Figure 1: Two major sources of tool error: radial shift and waviness

The two error sources cannot be completely separated, but both can be corrected simultaneously. When cutting a spherical shape, radial error in the tool flute location results in an error profile (departure from sphere) that has the shape of an “M” or “W” as shown in Figure 2. Tool waviness however results in an arbitrary error that will be unique to the individual tool. Here we show how to correct the tool path based on the effect of both error sources on a spherical artifact. We demonstrate the method for a specific geometry but it is

generalizable to any ball endmilling operation with a tool of any radius.

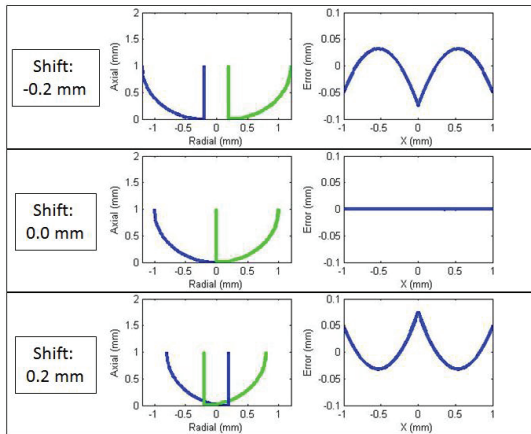


Figure 2: Effect of a radial shift of tool flute on departure from sphere error

EXPERIMENTAL ARRANGEMENT

The tool used in this study was a 1.5 mm single crystal single flute diamond endmill with zero rake angle. The artifact geometry selected to separate the tool errors was an 8 mm radius of curvature sphere with an aperture of 8 mm. This geometry produces a sphere with a 60° angular ($\pm 30^\circ$) sweep as shown in Figure 3 (a) below. For the case, where the spindle is aligned perpendicular to the XY-plane, 30° of the milling tool starting from the tip is used to generate the surface as shown in Figure 3 (b).

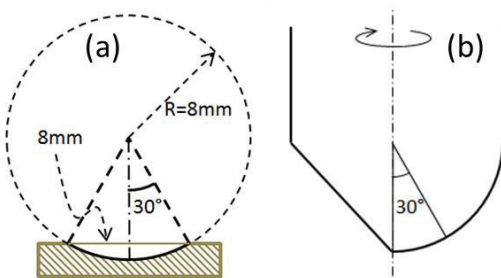


Figure 3: Artifact geometry (a) and the section of a ball mill measured by artifact (b)

The diamond milling machine used to conduct these experiments was a Moore Nanotechnology 350FG machine with two air bearing spindles, a 10,000 rpm turning spindle/rotary axis and a 60,000 rpm milling spindle. The milling procedure is a standard two-direction raster milling pattern with the

tool inclination angle α set to 0° or normal to the XY plane shown in Figure 4. The artifact was machined with a stepover of 0.020 mm, a feed rate of 200 mm/min, and a spindle speed of 40,000 rpm. The tool path was generated assuming a perfect tool with no radial shift error and no waviness. The tool path generation procedure is outlined below in the applications section.

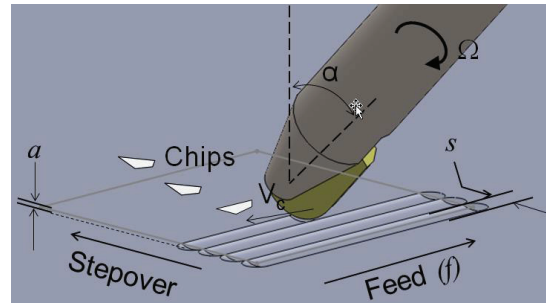


Figure 4: Raster milling with a ball end mill

The artifact was measured using a Zygo Verifire phase shifting interferometer with an F/0.75 transmission sphere. All tool path generation and data analysis was performed using Matlab 2013a.

EXPERIMENTAL RESULTS

The residual “M” or “W” shape comes after the best fit sphere has been subtracted from the measurement of the spherical artifact. A typical measurement of a spherical artifact cut without correction is shown in Figure 5 (right). A superimposed set of radial slices is shown in Figure 5 (left). This measurement of the uncorrected artifact is one half of the traditional “W” shaped error profile with tool waviness superimposed on it. Depending upon the relative amount of decentering error as compared to waviness error, the classic “M” or “W” error profile may be more or less evident.

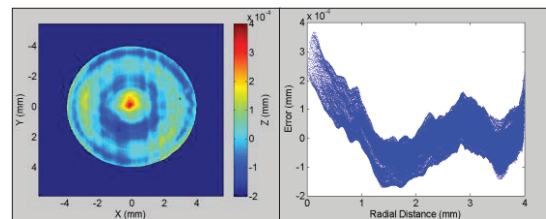


Figure 5: Uncorrected sphere measurement

As discussed below, the residual error showed a correctable form error with a peak-to-valley of about 300 nm. The

remaining error of about 150 nm PV is due to non-tool errors of the machining operation.

ANALYSIS

The process of correcting for the two dominant tool error sources is now described. Initially, the nominal spherical form is added back into the measured errors. The radial shift is determined by shifting the X and Y data radially inward and outward from the center of the data. After each shift, the best fit sphere is subtracted and the peak-to-valley error is determined. At some shift, the peak-to-valley error will reach a minimum as shown in Figure 6.

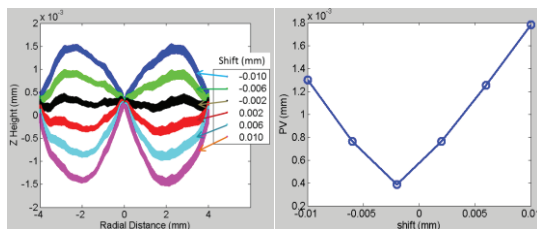


Figure 6: Example of radially shifting and removing best fit for a range of shifts (left), PV's of the residuals (right)

Radially shift the X and Y values and finding the peak-to-valley of the residual is an iterative process that ends when the variation in PV is on the same order as the surface finish requirements. The residual from the optimally shifted data becomes the starting data for the waviness correction simulation. The radial shift corrected data can be seen in Figure 7 represented by blue dots.

The remaining correctable error is rotationally invariant and can be mapped to the slope of the artifact and tool. The slope of the artifact is found at every point and the remaining error is fit to a polynomial. For the data shown here, an 8th order polynomial was used for the fitting shown in Figure 7 represented by a red line.

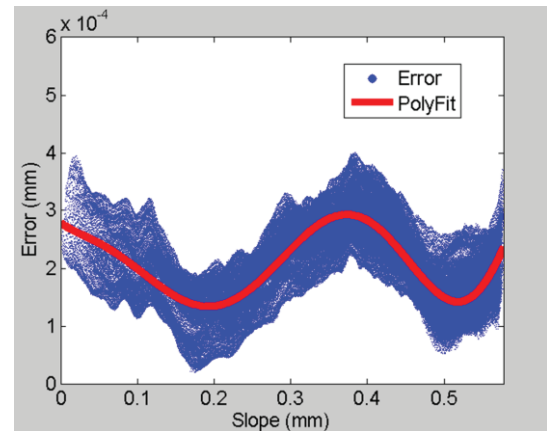


Figure 7: Artifact measurement with radial shift error removed and an 8th order polynomial fit

The uncorrectable error can be found by subtracting the polynomial fit from the data. The residual gives an estimate of the best possible error achievable in a final optic.

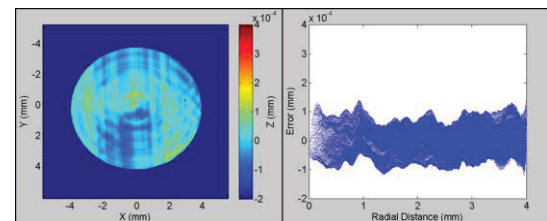


Figure 8: Fully tool corrected simulated measurement with the scale used in Figure 5

APPLICATION

The purpose of this work is to model the tool shape and correct the milling tool path of a freeform optic. The procedure developed to generate the corrected tool path starts with the assumption that the artifact was machined using a perfect non-corrected tool path. This assumption allows us to assume that the rotational invariant errors found on in the measurement came from tool errors. The theory behind generating the tool path for an optic using a spherical tool is that in order to cut a specific point on the optic (P_o) the vector created between the center point of the tool (P_t) and the point on the optic will be normal to the surface at point P_o . The tool path generation procedure is as follows:

1. Begin with the optical prescription described as a function of X and Y. Figure 9(a)
2. Find the partial derivatives as a function of X and Y.

3. Determine the points on the surface you wish the tool to contact.
4. Assemble the gradient vector at each point $\vec{\nabla}f = \langle -\frac{\partial f}{\partial X}, -\frac{\partial f}{\partial Y}, 1 \rangle$.
5. Compute normalized gradient by dividing the gradient by the magnitude of the vector at each point $\hat{n} = \langle n_x, n_y, n_z \rangle$. Figure 9(b)
6. Compute the radius compensation and correction by multiplying the unit normal by the tool radius of the tool, remembering that the tool radius is now a function of the slope of the tool.
7. Compute the normalized vector of the gradient only in the X-Y plane. This is the direction in which the radial shift correction will be applied.
8. Multiply the calculated radial shift correction by the vector found in step 7.
9. The final tool path is the sum of the points on the surface (step 3), the radius compensations (step 6), and the radial shift corrections (step 8). Figure 9(c,d)

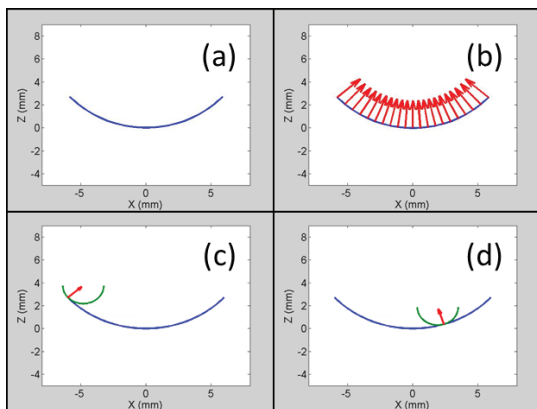


Figure 9: Tool compensation procedure simplified into 2 dimensions: (a) Surface prescription, (b) Unit normal, (c,d) Tool center shifted in direction of unit normal

This is the tool path generation procedure for a ball mill with spindle axis normal to the X-Y plane. If there spindle is oriented with an angle between the axis and the X-Y plane, then the radial shift correction will be applied in the plane that is normal to the axis.

DISCUSSION

Since the metrology of freeform optics is still a difficult task proving the effectiveness of this procedure has not been possible. The first optic we made included aspheric surfaces and could be used to take thermal images with only the lens and a detector. The optical design was provided by Dave Schmidt at Rochester Precision Optics. A measurement and image of one of the surfaces can be seen in Figure 10. The residual error was attributed to only correcting for radial shift and not waviness. The measurement was taken at OptiPro with their OptiTrace 5000.

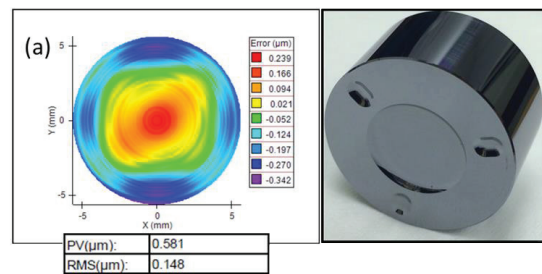


Figure 10: Aspheric Optic made after using this technique for only correcting radial error

The second optic made was a true freeform and used the full tool correction outlined above. The optic was a high order Alvarez lens designed by Shultz et al. [1]. The final optic and designed prescription surface map can be seen in Figure 11. On the optics substrate we also machined three metrology fiducials that would allow the measurement of the optic to be oriented with the proper coordinate system to compare with the prescription.

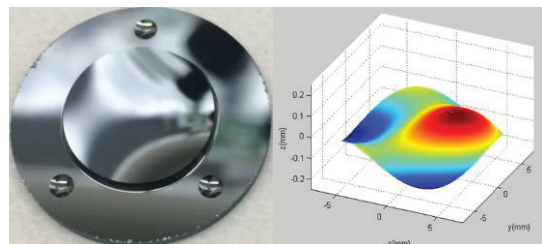


Figure 11: Machined Alvarez optic with metrology fiducials (left), surface map (right)

The plan for this optic was to functionally test the Alvarez the same way outlined by P. Smilie in 2012 [2]. Functional testing was performed by Barnhardt et al. and will be reported in a later publication. However,

they found good correlation between the theoretical and experimental measurements. This functional testing serves to qualitatively confirm the form error of the optic.

REFERENCES

- [1] Shultz, J., Suleski, T., Design of a Higher Order Alvarez Lens, 2015, In preparation.
- [2] P.J. Smilie, B.S. Dutterer, J.L. Lineberger, M.A. Davies and T.J. Suleski, Design and characterization of an infrared Alvarez lens, Opt. Eng. 51(1), 013006, 2012
- [3] Barnhardt, D., Owen, J., Shultz, J., Davies, M., Suleski, T. Design, Manufacture, and Testing of Higher Order Alvarez Lens in IRG 26, 2015, In preparation.

ACKNOWLEDGEMENTS

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3 AXIS HOMODYNE DISPLACEMENT MEASURING INTERFEROMETER PROBE FOR FREEFORM OPTICS

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INTRODUCTION

Modern computer-controlled optical fabrication techniques have enabled optical designers to explore new avenues for optical design, particularly in freeform optics. However, further advancement in metrology techniques is required to fully realize the capabilities of freeform optics and systems built using these components.

Freeform optics present a unique challenge to optical manufacturers because of their asymmetry, complex forms, potentially high slopes, and steep departures from spherical surfaces. A lack of reliable metrology is one reason manufacturers have difficulty with freeforms, as traditional, high-resolution metrology systems typically cannot cope with such complex surfaces. This shortcoming has created a need for advanced metrology techniques. Our slope-sensitive optical measuring device, which is capable of resolving the intricate shapes of a freeform optic, is intended to address this shortcoming.

In this research, a novel 3 degree of freedom (DoF) displacement measuring interferometer probe is designed and results to date are shown. The desired specifications for the final system are sub-10 nm linear displacement and sub-5 μ rad angular rotation resolution, along with a 1 kHz measurement bandwidth and a 15 μ m measurement spot size. The probe will be mounted on an optical coordinate measuring machine, specifically the OptiPro UltraSurf 5x [1].

METHODS

Our system, shown in Figure 1, employs a homodyne interferometer in a Michelson configuration for linear displacement measurement, and a 2-axis position sensing detector (PSD) for angular measurement using a standard autocollimation technique. The homodyne detection method uses two photodetectors and an induced $\pi/4$ radian phase shift to retain directional sensitivity. The arctangent of the two sinusoidal signals is

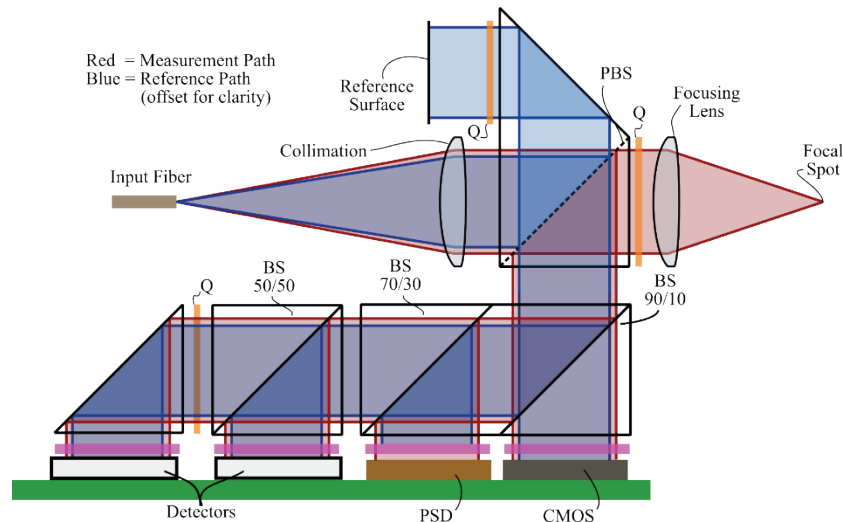


FIGURE 1. System schematic. Michelson interferometer configuration, polarization optics, PSD and photodetectors, and CMOS sensor for visual confirmation of focus on measurement surface are indicated.

computed and unwrapped, resulting in a displacement measurement based on the wavelength of the 633 nm HeNe laser source. The angular measurement uses a PSD to sense the linear displacement of the centroid of the laser induced by angular displacement at the measurement surface due to the focusing lens included in the measurement arm. This relationship is described by

$$\alpha = \frac{1}{2} \arctan\left(\frac{\delta y}{f}\right), \quad (1),$$

where α is the slope angle, δy is the displacement of the beam centroid, and f is the focal length of the measurement lens. Figure 2 illustrates the relationship between angular displacement at the measurement surface and linear displacement on the PSD. A polarizer is used to block undesirable light from the reference arm of the interferometer [2].

To measure a surface, the probe will follow a point-to-point measurement procedure. An operator will initially focus the probe to a known

offset using the CMOS sensor, and also set the probe normal to the measurement surface using the PSD. The probe will move a desired

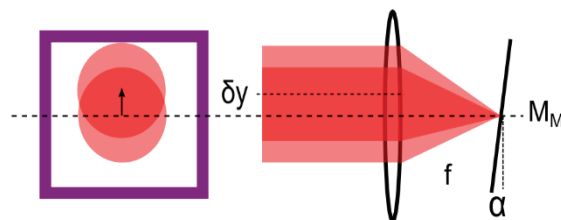


FIGURE 2. Angular displacement methodology. Angular displacement at measurement surface induces linear displacement on PSD.

increment along the measurement surface and record the relative change in slope and offset. Using this measurement data, the OCMM repositions itself to correct the probe back to the original offset and normal vector to the surface. This process, as shown in Figure 3, is repeated until the entire surface has been scanned. Thus, the overall surface profile is determined by the OCMM positioning system, optical probe, and the nominal offset distance.

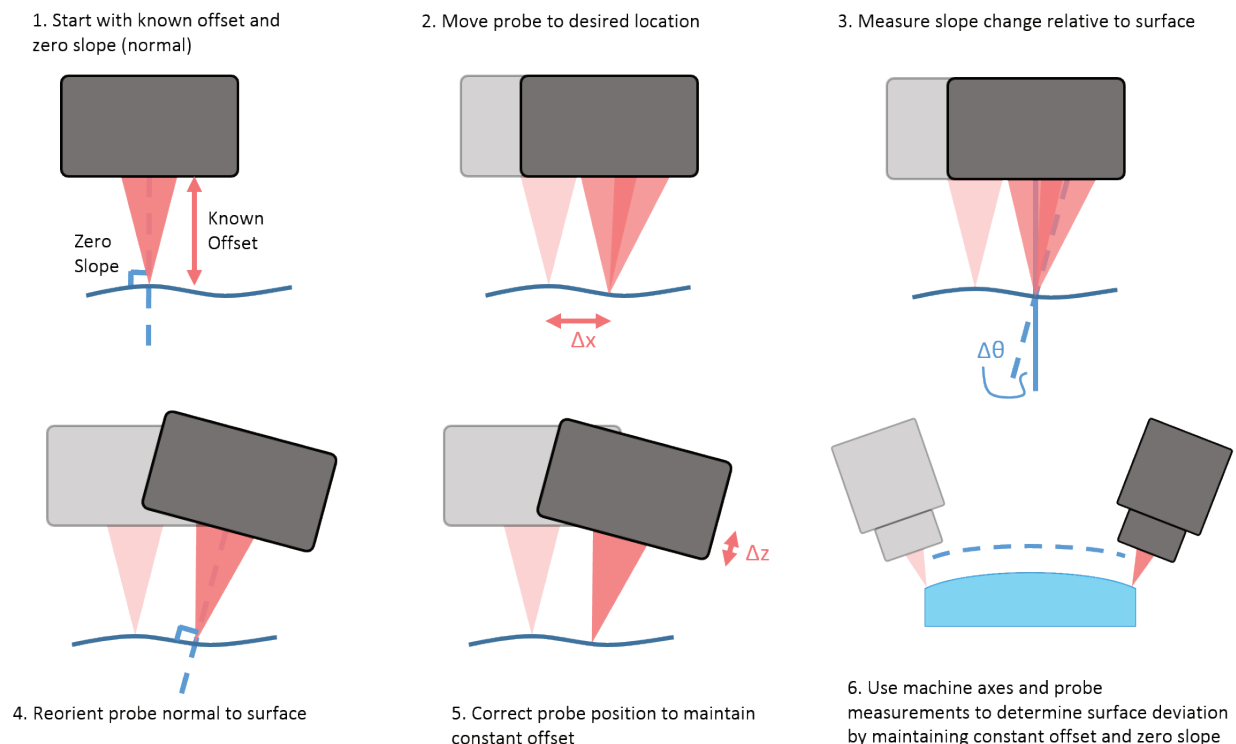


FIGURE 3. Schematic of point to point probing method once integrated to OptiPro UltraSurf 5X. Probe will remain normal to and at set displacement to measurement surface using closed loop feedback. The OCMM maintains a constant offset with respect to the part surface, using the probe to ensure the measurement is performed normal to the surface.

DESIGN

The main requirement of the homodyne probe mechanical design was to have a compact, and lightweight design that could be readily integrated into the OptiPro UltraSurf 5X CMM platform. A small footprint would allow the probe to be used on freeform surfaces with limited space constraints. Keeping the probe as lightweight as possible will allow the UltraSurf's measurement arm to handle the probe without deflection and measure with a higher bandwidth. A lighter probe will minimize dynamic effects as well.

Other requirements include a fiber-delivered source to isolate the thermally-sensitive optics from the heat of the laser, onboard optical sensors to retain signal fidelity, and a very fine, precise alignment mechanism that would be able to withstand the working forces of an OCMM measurement.

The culmination of these design criteria can be seen in the CAD models of the homodyne probe in Figure 4 and Figure 5. Because of the homodyne detection method, the probe is thermally stable, due to very small non-common optical path lengths between the measurement and reference arms. The probe is made from Aluminum 6061, with a base body, and 1 wall for the input mechanism, and a lightweight plastic top and bottom cover. The exterior dimensions of the probe body are 2.0 in. x 1.75 in. x 1.7 in.

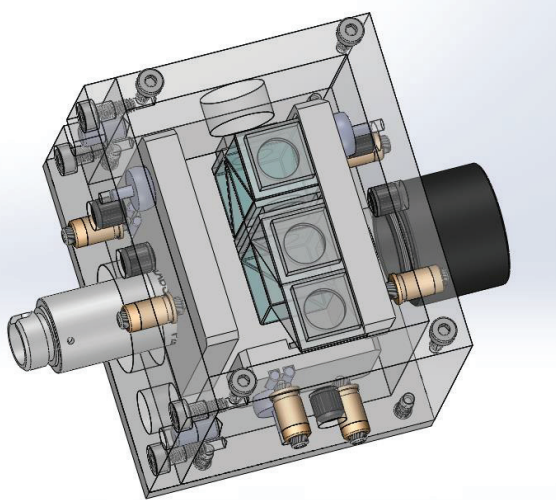


FIGURE 4. CAD image of the 3-axis probe with the cover and side wall transparent to reveal the optics and their mounting mechanisms.

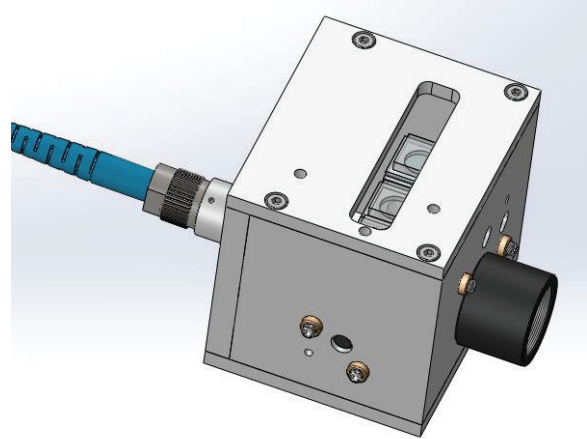


FIGURE 5. CAD image of 3-axis probe completely assembled. Clearance slot for the optical sensors can be seen on the top cover.

Inside the probe is one custom mount for 10 mm beamsplitters, and two custom mounts for polarization optics (polarizers and quarter wave plates). In addition to custom mounts, the design uses custom cut polarizers and quarter wave plates to enable the probe to remain as compact as possible. Stock optics were cut down to sizes of 8 x 8 mm for the polarizers, and 8 x 10 mm for the quarter wave plates. The design features a fiber delivered input source from a polarization maintaining FC/APC fiber patch cable. The input fiber is oriented to a polarization of 45° and connected to a Thorlabs FC/APC fixed focus fiber collimation package. The collimation package is threaded into an internal mount which has pitch and yaw adjustments using a Kelvin clamp configuration of a cone, V-groove, and flat as shown in Figure 6. A high-pull neodymium magnet is used to constrain the mount to the probe body. Fine pitch M2.5 x 0.2 adjustment screws allow for 200 μm of travel per revolution, providing for a more precise and user-friendly alignment procedure with finer pitch screws than typical kinematic mounts. The input wall is assembled onto slip fit dowel pins which allow for small amounts of translational X and Y alignment, giving the input assembly a total of 4 DoF.

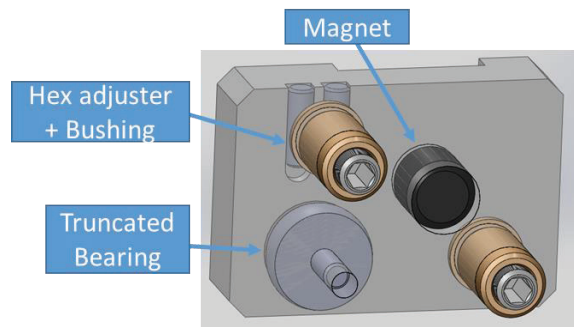


FIGURE 6. Kelvin clamp mechanism using two fine pitch adjustment screws in a V-groove and flat which tip and tilt the mount about a ball bearing pivoting from a cone. The mount position is held in place by a magnet.

Following the input optics, the collimated light passes through a series of beamsplitters which have been conjoined by epoxying custom cut 1 mm thick microscope slides between beamsplitters, as well as between polarizers and wave plates which are also attached to the stack, as shown in Figure 7. These microscope slides allow for a one piece combined optic which eliminates extra alignment mechanisms and further limits the size of the probe. The microscope slides are also important to provide a buffer to prevent interference between adjacent AR coated faces of the optics. Additionally, the microscope slides also are constructed of glass which will expand similarly to the optics they are holding together, helping them remain thermally stable. The beamsplitter stack also has pitch and yaw adjustments using a Kelvin clamp configuration as explained above. It is important to have these degrees of freedom in order to ensure that the measurement and reference arms are collinear and aligned to the center of the photodiode and PSD sensors.

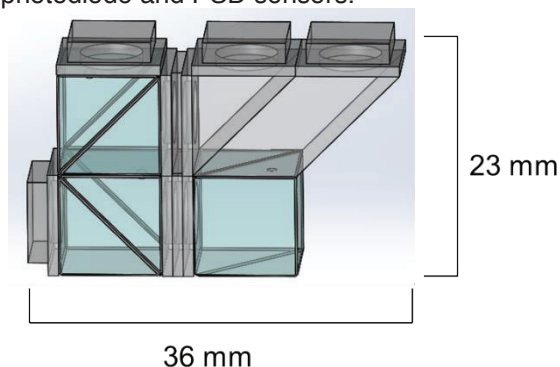


FIGURE 7. A one-piece beamsplitter stack allows for compact size of the probe. Microscope slides are used to conjoin the optics with epoxy.

The reference surface has been combined into the reference quarter wave plate by adding a reflective coating to the rear face of the quarter wave plate which takes the place of an additional mirror. The reference quarter wave plate and the measurement quarter wave plate are mounted vertically in a loosely-toleranced pocket to allow for small adjustments to the fast axis before securing the wave plate in place with epoxy. Similar to the primary polarizing beamsplitter, the reference surface also has pitch and yaw adjustment provided by a Kelvin clamp.

The measurement arm of the probe features a stock Thorlabs 0.5 inch lens mount and a 50 mm focal length bi-convex lens. The smallest slope angle that can be resolved by the PSD is inversely proportional to the focal length, as shown in equation 1. A 50 mm focal length is sufficiently long to allow for the desired slope sensitivity while also not requiring the probe to stand far from the surface, which could increase errors caused by air currents in free space.

Once the light has recombined after reflecting off of the measurement and reference surfaces, it travels through a 90/10 beam splitter, which siphons off a small amount of the light to a CMOS sensor. The CMOS sensor allows the machine operator to ensure the probe is focused on the measurement surface prior to beginning a measurement run by examining the interference fringes. The remaining light travels through the remaining beamsplitters and polarization optics which are mounted on the same plane in order to increase ease of alignment. Photodiodes and the PSD are mounted along the side of the probe.

A custom circuit board, as shown in Figure 8, has been designed for on-board detection and filtering of the signal. Each photodiode generates a current signal that is converted to a voltage signal via a transimpedance amplifier designed as a low-pass amplifier for signals up to 30 kHz. A second-generation device will be tuned to 100 kHz. These signals are then processed in Matlab to provide displacement measurements with directional sensitivity.

The PSD provides four voltage signals, each of which is passed through a Sallen-Key low-pass filter design to attenuate noise above 100 kHz.

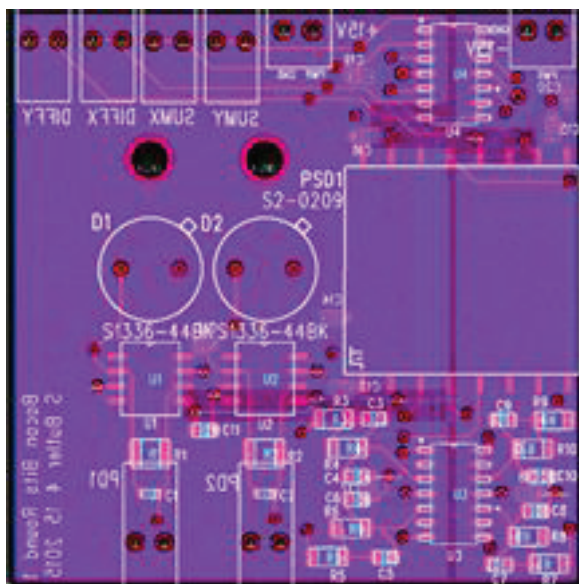


FIGURE 8. Custom circuit board layout for onboard signal acquisition and filtering.

These four signals are used to calculate the centroid of the beam via simple algebraic equations. Equation 1 is then used to calculate the slope at the measurement surface.

TEST SETUP

A benchtop setup of the probe optics and electronics was constructed in a lab environment. To verify the linear resolution of the probe, a mirror was mounted to a New Focus Picomotor 5-axis piezostepper stage. Measurements were taken by stepping the piezo stage $10\ \mu\text{m}$ at a rate of 100 Hz. The probe displacement data was verified against the industry standard Renishaw Displacement Measuring Interferometer.

Voltage data from the displacement measuring photodetectors is collected and processed in Matlab using the atan2 function. This processed information is comprised of periodic discontinuous segments which were then unwrapped and normalized. Finally, the normalized data is converted to nanometer displacements by multiplying by the wavelength of 633 nm, and dividing by the number of passes, which is 2 for the homodyne probe.

RESULTS

Benchtop results indicate a 10 nm noise level for linear displacement measurement and a directionally-dependent noise level of $20\ \mu\text{rad}$ and $100\ \mu\text{rad}$ in θ_x and θ_y . The linear noise level is dominated by periodic error caused by polarization mixing. Free-space testing using the

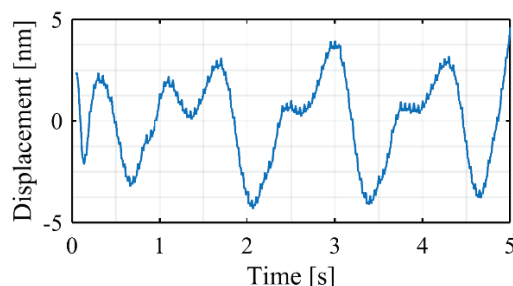


FIGURE 9. Uncompensated data retains periodic error from polarization mixing and still achieves sub-10 nm noise level. A 90 Hz low-pass filter removes parasitic stage motion information.

custom electronics board and a 90 Hz filter to eliminate parasitic stage motion resulted in the noise floor data shown in Figure 9.

A sub-nanometer noise level is possible using a Kalman filter-based compensation algorithm, as seen in Figure 10. Real-time implementation of this algorithm is ongoing. The angular noise level is suspected to be an artifact of the free space mounting, and is expected to improve beyond the results shown in Figure 11 with implementation in the actual probe.

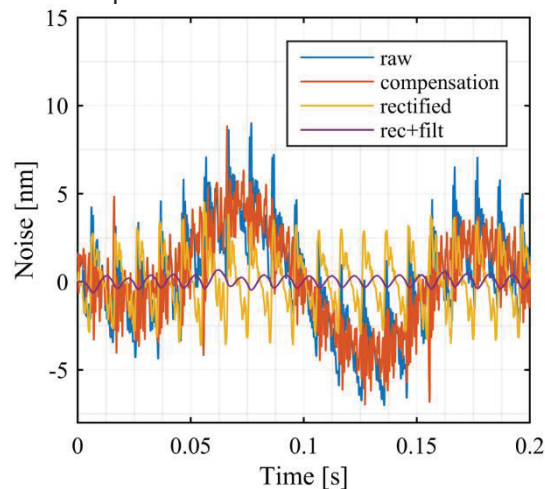


FIGURE 10. Raw data containing low-order periodic error and matching “compensation” data generated to fit raw data. Rectified data has had compensation data removed from the raw data, while rec+filt data has had a 90 Hz filter applied to remove parasitic stage motion information.

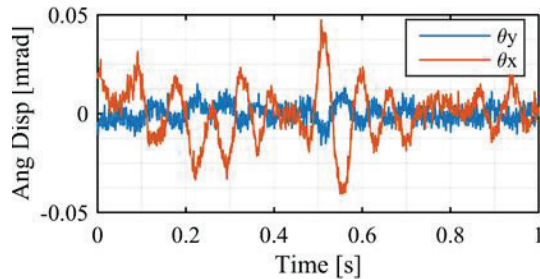


FIGURE 11. Baseline angular noise level is directionally dependent, suggesting mounting scheme is affecting performance.

FUTURE WORK

The probe will be tested against the industry standard Renishaw Displacement Measuring interferometer, using a 5 axis piezoelectric stage for comparison of linear displacement. The probe will be integrated to the OptiPro UltraSurf CMM platform to test a freeform surface such as the Alvarez lens, and compare the data to existing freeform metrology methods.

ACKNOWLEDGMENTS

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REFERENCES

- [1] Fang FZ, Zhang XD, Weckenmann A, Zhang GX, Evans C. Manufacturing and measurement of freeform optics. CIRP Annals – Manufacturing Technology 62(2). 823-846. 2013.
- [2] DeFisher S, Bechtold M, Mohring D. A non-contact surface measurement system for freeform and conformal optics. Proc. SPIE 8016. 2011.
- [3] Harding, K. Handbook of Optical Dimensional Metrology. CRC Press. Boca Raton, Florida. 2013.
- [4] Joo KN, Ellis J, Spronck J, van Kan P, Munnig Schmidt R, Simple heterodyne laser interferometer with subnanometer periodic errors. Opt. Lett. 34. 386-388. 2009.

Manufacturing polymeric micro lenses and lens arrays by a multiphase printing method

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INSTRUCTION

Micro lenses and lens arrays are widely studied in the past decades. They can find applications in light beam shape, imaging, sensing, micro lithography and so on [1-3]. Numerous manufacturing methods have been developed in order to fabricate micro lenses and lens arrays, such as lithography (laser lithography and photolithography) [4,5], diamond turning [6], ultraprecision micro-milling [7], laser machining [8-10], molding (injection molding and hot embossing) [11-13], electro based methods (electro-wetting, electro-dewetting and electro-pulling)[14-16], thermal reflow [17,18], chemical etching[19], inkjet printing[20-22], and multiphase method[23-26]. These methods have their own advantages and disadvantages. For example, diamond turning and ultraprecision micro-milling are able to form lenses and lens-arrays with freeform profiles. However, they require sophisticated facilities. Electro based methods can form parabolic convex micro lenses by using electrostatic forces. This method needs electrostatic power supply as well as controllers in order to form electro field and it makes the process more complicated. Thermal reflow and chemical etching enable wafer-level fabrication, but they only fabricate one side micro lenses and these methods need pre-procedure, such as photolithography. Inkjet printing is studied intensively in recent years. This printing method has been used to directly print micro to meso scale monomer drops on solid substrates and followed by UV-induced polymerization to form solid polymer lenses and lens arrays. Polymeric lenses can automatically formed by surface tension with a good optical surface finish. Compared to other method, manufacturing process of inkjet printing is simple, quick, and low cost. But the lens by using this method is Plano-convex and the profile is determined by the contact angle of the polymer

material and the substrate. Surface treatment is applied on the solid substrate in order to tune the lens profile. However, extra procedures and facilities need be involved. Multiphase method is also mainly based on the surface tension, but it can form biconvex micro lens due to two fluid-fluid interface or fluid-wall interface. Lens profile can be tuned by changing the interface tension in this method without extra steps and facilities. It is a low cost method to fabricate polymeric micro and meso lenses with different profiles. We have developed biconvex micro lens and lens arrays using a multiphase method [24]. We also used micro-fluidic method to generate small monomer drops with different viscosity, and then use a multiphase method to form spherical and drum shape micro lenses. In this abstract, the multiphase method will be introduced and discussed.

METHOD

Two different cases that utilizes the multiphase method will be discussed in this abstract. In the first case, biconvex micro lenses were formed in fluid-fluid condition. In the second case, Plano-convex micro lenses were formed on substrate with a liquid environment.

Biconvex micro lens (case 1)

In this case, electrostatic printing method was used to generate monomer droplets. Monomer droplets were deposited on a liquid surface. Partial wetting phenomenon [27-29] is used to fabricate biconvex parabolic lenses (Fig. 1a). Fig. 1b schematically illustrates the force balance of a droplet on a liquid surface. To be able to float on the liquid surface, the gravity of the pre-polymer lens, $\vec{G}$, needs be balance the buoyant force, $\vec{F}_b$, and surface tension force, $\vec{F}_s$ (Fig 1b).

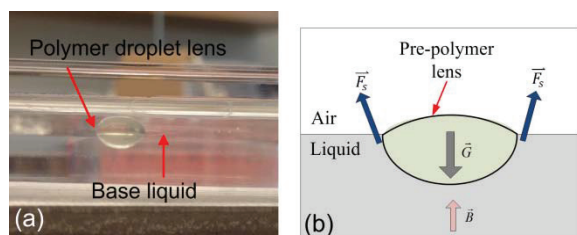


Fig. 1. (a) Monomer liquid lens formed on a liquid surface. (b) Force balance of a monomer droplet on the liquid surface.

In this work, we used a mixture of two monomer materials, i.e. Triallyl-1,3,5-triazine-2,4,6-trione (TATATO) (Sigma-Aldrich) and Pentaerythritol Tetra(3-mercaptopropionate) (PETMP) (Sigma-Aldrich). Irgacure 754 (Ir754, PTHutchins) was used as a photo initiator. The basic properties of the materials are shown in Table 1. Base liquid was deionized water with surfactant (detergent, Brand: Dawn). The surfactant was used to adjust the interfacial tension between monomer droplets and the base liquid and thus adjust the shape of the lenses. Liquid lenses were cured by using a UV Lamp.

Table 1 Material properties

Composition	Density g/cm ³	Refractive Index	Ratio (wt %)
TATATO	1.159	1.510	0.6
PETMP	1.3±0.1	1.531	0.4
Ir754	--	--	0.0015

Plano-convex micro lens (case 2)

In this case, droplets were generated by using microfluidic dispensers (dimensions in Table 2) and collected on a substrate (Fig. 2a). A reservoir wall, which was made from PDMS, was used to control the height of the water phase (Fig. 2b and d). Extra water flowed over the reservoir wall and was removed, while monomer droplets were stay within the reservoir. Since the monomer materials have larger densities than water, their droplets sank down to the substrate and are surrounded by water. A LED UV lamp (365nm, CS2010, THORLABS) was then used to solidify monomer lenses. As can be seen from Fig. 2 c and e, with different water phase heights, the obtained lenses had different shapes. Fig. 2c shows the lens formed when partially surrounded by water. Fig. 2e shows the lens formed when fully surrounded by water.

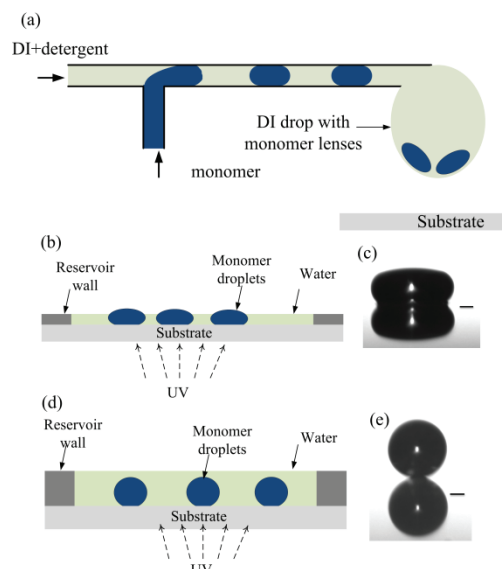


Fig. 2. (a) generating monomer droplets by a microfluidic dispenser, (b) lens forming by insufficient water, (c) lens formed with insufficient water, (d) set up of lens forming by sufficient water, and (e) lens formed with sufficient water. (Scale bar: 100μm).

Three UV curable optical adhesives, NOA72, NOA76, NOA68T (Norland Products) were used as lens materials. Deionized (DI) water with surfactant (detergent, concentration 5ml/l, V/V) was used as the surrounding fluid. Some physical properties of these materials are listed in Table 3.

Table 2. Microfluidic dispenser dimensions

Device	1	2	3	4
Main Channel(mm)	0.51	0.64	1.59	0.64
Vertical Channel(mm)	0.1	0.15	0.51	0.51

Table 3. Material properties

Material	NOA 72	NOA 76	NOA 68T	DI water with surfactant
Viscosity at 25 °C (cps)	155	4500	22000	~0.890*
Surface tension (mPa s)	41.85	41.35	41.9	24.6
Specific	1.2	1.12	1.2	~1**

gravity				
Refractive Index after curing	1.56	1.51	1.54	NA

* viscosity and specific gravity of DI water with surfactant are approximated by using the data of DI water.

RESULT

The lenses profiles, and optical properties were tested and analyzed for both two cases.

Biconvex micro lens (case 1)

As can be seen from Figure 3, both the top (formed in air phase) and bottom (formed in the base liquid) surfaces have curved shapes which can be approximated as parabolas. Due to the difference in surface tension affecting the lens surfaces, the parabolic fit coefficients describing the top and bottom profiles are different.

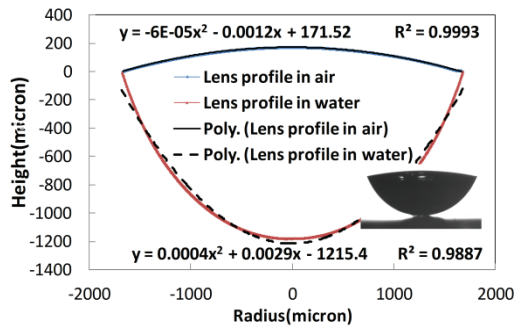


Fig. 3. Profile and the fitting curves of a lenslet that formed in a base liquid with a surfactant concentration of 8×10^{-6} (insert shows the picture of the lens).

In this research, the influence of surface tension was investigated. Different detergent concentrations, from 5×10^{-6} to 10×10^{-6} V/V, were used. The relationship of lenslet shape and surfactant concentration can be seen in Fig. 4a and b. With the increase of surfactant, the surface tension between the monomer and water decreases, smaller curvatures (Fig. 4a) and larger height-to-diameter (H/D) ratios (Fig. 4b) were obtained. The results in Figure 3 and 4 provide us guidelines to find the appropriate conditions in order to fabricate lenslets with designed shape.

Fig. 5a shows an 8 by 8 micro lens array manually assembled on a Polydimethylsiloxane (PDMS) substrate by using formed lenslets which have 2mm aperture. Fig. 6b shows WSU Cougar Logo formed by using the lens array.

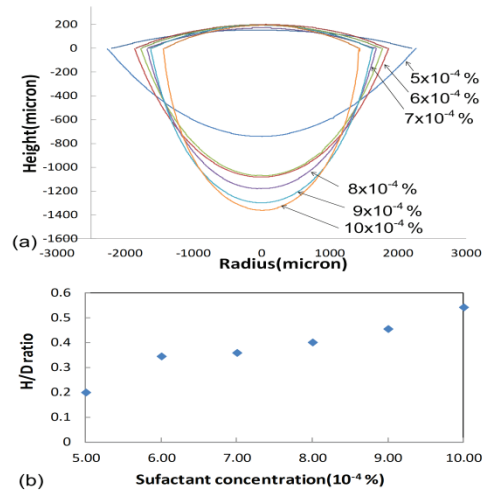


Fig. 4. (a) Relationship of lenslet shape and surfactant concentration. (b) Relationship of lenslet height-to-diameter (H/D) ratio and surfactant concentration.

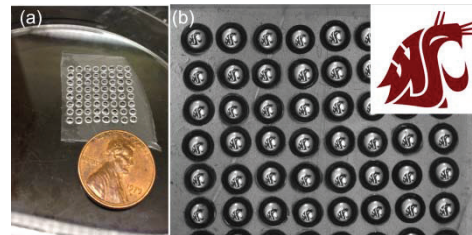


Fig. 5. (a) An assembled 8 by 8 micro lens array on a PDMS substrate. (b) WSU Cougar Logo formed by using the lens array, (the insert shows the Cougar logo).

Fluid-fluid-wall condition (case 2)

Fig. 6 shows profiles of the lenses by using different materials under different forming conditions. In Fig. 6, experiment condition was expressed by 3 numbers and one letter. For instance, 100-1-2-S means the flow rate of the continuous phase in microfluidic device was $100 \mu\text{l}/\text{min}$, the flow rate of the discontinuous phase in microfluidic device was $1 \mu\text{l}/\text{min}$, device 2, and letter S means the droplet was fully covered with sufficient water. Letter I means the droplet was partially covered by insufficient water. For the condition of sufficient water, the lens shape is close to a sphere with a small flat area at the bottom. For the condition of insufficient water, there is a flat area at the bottom, which is larger than the bottom area of the lens formed in sufficient water condition. The lens formed by insufficient water is close to a drum shape. From Fig. 6, the lenses formed in

sufficient water condition had the similar spherical shape, and the difference is the size. For the lens in insufficient water condition, they all had similar drum shapes. From Fig. 6, the viscosity of the monomer material had little effect on the lens profile, and the level of surrounding water was the main factor. This is because all the lenses in Fig. 6 were solidified after they reached equilibrium, while viscosity mainly affects dynamic process.

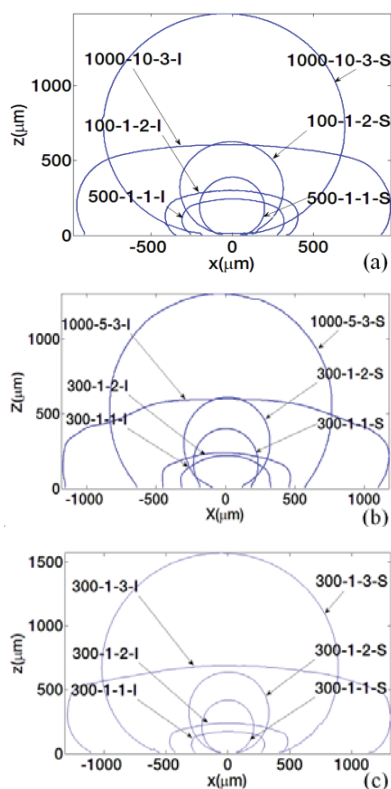


Fig. 6. Profiles of lenses of different materials, (a) NOA 72, (b) NOA 76, (c) NOA 68T.

This method is also able to fabricate lens arrays by self-organization. In our method, generated in microfluidic devices, droplets were surrounded with water/surfactant and were coated with a thin layer of water and surfactant. With this coating, monomer lenses were able to touch other lenses without coalescing to a large one. During the test, monomer lenses coalesced into a large lens when we used DI water only. After water and small monomer droplets falling on the substrate, water starts to recede for a minimal surface area. At the same time, due to the monomer droplets existed inside, extra interface areas were created. To minimize the interface area, small monomer droplets tend to closely packed to reduce the interface area and thus self-organized into a lens array. Fig 7 shows a

self-organized micro lens array fabricated by using microfluidic device 4. The lens array has high fill factor 88.7%. The coefficient of variation (ratio of standard deviation of aperture size to mean aperture size) is 0.0256.

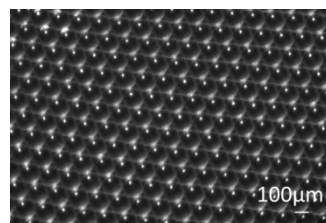
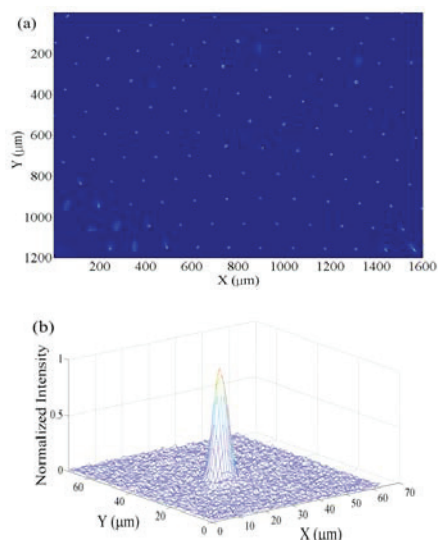


Fig. 7. Self-assembled micro lens array with 130 μ m aperture size (the white dots were the images of the illumination light),

The effective focal lengths of two typical lenses of same volume and different shapes (1000-10-3-S and 1000-10-3-I) were measured. It is 1.53 mm for the lens formed by sufficient water (spherical shape) and is 7.16 mm for the lens formed by insufficient water (drum shape).

Fig. 8a shows the point spread function (PSF) of a self-assembled lens array by using a He-Ne laser and a 25 μ m aperture as the point light source. Fig. 8b shows the details of a PSF of one of the lens. Fig. 8c shows WSU Cougar Logo images formed by using another lens array with 1mm aperture size. As evident from Fig. 7 and Fig.8, uniform lens arrays, with well-order, high fill factor and good imaging quality can be generated using our method.



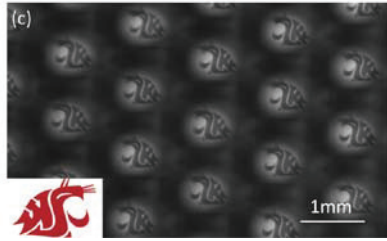


Fig. 8, (a) Point spread function of a self-organized micro lens array, (b) point spread function of a single microlens, (c) WSU cougar logo formed by using a lens array with 1mm aperture size (the insert shows the cougar logo).

CONCLUSION

This work reported a low cost multiphase printing method to fabricate polymeric lenses and lens arrays by using interface forces, which is able to create micro to meso scale lenses with free forms. Both the lenslet size and shape can be controlled by adjusting manufacturing parameters. This method can also fabricate self-organized lens arrays. The formed lenslets have good surface finish as well as good imaging quality and high fill factor. For future research, quantitative models for lens shape and self-organization mechanisms will be expected.

REFERENCES

- [1] Z. Deng, Q. Yang, F. Chen, H. Bian, J. Yong, G. Du, Y. Hu, and X. Hou, "High-Performance Laser Beam Homogenizer Based on Double-Sided Concave Microlens," *IEEE Photonics Technol. Lett.*, vol. 26, no. 20, pp. 2086–2089, Oct. 2014.
- [2] P. Nussbaum, R. Völkel, H. P. Herzig, M. Eisner, and S. Haselbeck, "Design, fabrication and testing of microlens arrays for sensors and microsystems," *Pure Appl. Opt. J. Eur. Opt. Soc. Part A*, vol. 6, no. 6, p. 617, Nov. 1997.
- [3] Y.-K. Ee, P. Kumnorkaew, R. A. Arif, H. Tong, J. F. Gilchrist, and N. Tansu, "Light extraction efficiency enhancement of InGaN quantum wells light-emitting diodes with polydimethylsiloxane concave microstructures," *Opt. Express*, vol. 17, no. 16, pp. 13747–13757, Aug. 2009.
- [4] D. Radtke, J. Duparré, U. D. Zeitner, and A. Tünnermann, "Laser lithographic fabrication and characterization of a spherical artificial compound eye," *Opt. Express*, vol. 15, no. 6, pp. 3067–3077, Mar. 2007.
- [5] "SPIE | Optical Engineering | Microlens array fabricated by a low-cost grayscale lithography maskless system." [Online]. Available: <http://opticalengineering.spiedigitallibrary.org/article.aspx?articleid=1790555>. [Accessed: 28-Aug-2015].
- [6] S. Scheiding, A. Y. Yi, A. Gebhardt, L. Li, S. Risse, R. Eberhardt, and A. Tünnermann, "Freeform manufacturing of a microoptical lens array on a steep curved substrate by use of a voice coil fast tool servo," *Opt. Express*, vol. 19, no. 24, p. 23938, Nov. 2011.
- [7] S. Scheiding, A. Y. Yi, A. Gebhardt, R. Loose, L. Li, S. Risse, R. Eberhardt, and A. Tünnermann, "Diamond milling or turning for the fabrication of micro lens arrays: comparing different diamond machining technologies," 2011, vol. 7927, p. 79270N–79270N–11.
- [8] J. Shao, Y. Ding, H. Zhai, B. Hu, X. Li, and H. Tian, "Fabrication of large curvature microlens array using confined laser swelling method," *Opt. Lett.*, vol. 38, no. 16, pp. 3044–3046, Aug. 2013.
- [9] C. Zheng, A. Hu, K. D. Kihm, Q. Ma, R. Li, T. Chen, and W. W. Duley, "Femtosecond Laser Fabrication of Cavity Microball Lens (CMBL) inside a PMMA Substrate for Super-Wide Angle Imaging," *Small*, p. n/a–n/a, Mar. 2015.
- [10] M. Malinauskas, M. Farsari, A. Piskarskas, and S. Juodkakis, "Ultrafast laser nanostructuring of photopolymers: A decade of advances," *Phys. Rep.*, vol. 533, no. 1, pp. 1–31, Dec. 2013.
- [11] S.-W. Tsai and Y.-C. Lee, "Fabrication of ball-strip convex microlens array using seamless roller mold patterned by curved surface lithography technique," *J. Micromechanics Microengineering*, vol. 24, no. 1, p. 015014, Jan. 2014.
- [12] "Manufacturing of a precision 3D microlens array on a steep curved substrate by injection molding process : Advanced Optical Technologies." [Online]. Available: <http://www.degruyter.com/view/j/aot.2013.2.issue-3/aot-2012-0061/aot-2012-0061.xml>. [Accessed: 28-Aug-2015].
- [13] P. Xie, P. He, Y.-C. Yen, K. J. Kwak, D. Gallego-Perez, L. Chang, W. Liao, A. Yi, and L. J. Lee, "Rapid hot embossing of polymer microstructures using carbide-bonded graphene coating on silicon

- stampers," *Surf. Coat. Technol.*, vol. 258, pp. 174–180, Nov. 2014.
- [14] S. Grilli, L. Miccio, V. Vespini, A. Finizio, S. De Nicola, and P. Ferraro, "Liquid microlens array activated by selective electrowetting on lithium niobate substrates," *Opt. Express*, vol. 16, no. 11, pp. 8084–8093, May 2008.
- [15] X. Li, H. Tian, Y. Ding, J. Shao, and Y. Wei, "Electrically Templated Dewetting of a UV-Curable Prepolymer Film for the Fabrication of a Concave Microlens Array with Well-Defined Curvature," *ACS Appl. Mater. Interfaces*, vol. 5, no. 20, pp. 9975–9982, Oct. 2013.
- [16] H.-C. Wei and G.-D. J. Su, "Fabrication of a transparent and self-assembled microlens array using hydrophilic effect and electric field pulling," *J. Micromechanics Microengineering*, vol. 22, no. 2, p. 025007, Feb. 2012.
- [17] H. Zhang, L. Li, D. L. McCray, D. Yao, and A. Y. Yi, "A microlens array on curved substrates by 3D micro projection and reflow process," *Sens. Actuators Phys.*, vol. 179, pp. 242–250, Jun. 2012.
- [18] "High fill-factor microlens array mold insert fabrication using a thermal reflow process - IOPscience." [Online]. Available: <http://iopscience.iop.org/article/10.1088/0960-1317/14/8/012/meta;jsessionid=E3EF459676A1D6993A01D432A6FB8D81.c1>. [Accessed: 28-Aug-2015].
- [19] F. Chen, H. Liu, Q. Yang, X. Wang, C. Hou, H. Bian, W. Liang, J. Si, and X. Hou, "Maskless fabrication of concave microlens arrays on silica glasses by a femtosecond-laser-enhanced local wet etching method," *Opt. Express*, vol. 18, no. 19, pp. 20334–20343, Sep. 2010.
- [20] J.-P. Lu, W.-K. Huang, and F.-C. Chen, "Self-positioning microlens arrays prepared using ink-jet printing," *Opt. Eng.*, vol. 48, no. 7, pp. 073606–073606–3, 2009.
- [21] J. Y. Kim, K. Pfeiffer, A. Voigt, G. Gruetzner, and J. Brugger, "Directly fabricated multi-scale microlens arrays on a hydrophobic flat surface by a simple ink-jet printing technique," *J. Mater. Chem.*, vol. 22, no. 7, p. 3053, 2012.
- [22] J. Y. Kim, N. B. Brauer, V. Fakhfour, D. L. Boiko, E. Charbon, G. Gruetzner, and J. Brugger, "Hybrid polymer microlens arrays with high numerical apertures fabricated using simple ink-jet printing technique," *Opt. Mater. Express*, vol. 1, no. 2, p. 259, Jun. 2011.
- [23] S.-Y. Hsiao, C.-C. Lee, and W. Fang, "Novel concave-based micro optical components," in *IEEE 21st International Conference on Micro Electro Mechanical Systems, 2008. MEMS 2008*, 2008, pp. 124–127.
- [24] R. Sun, Y. Li, and L. Li, "Rapid method for fabricating polymeric biconvex parabolic lenslets," *Opt. Lett.*, vol. 39, no. 18, p. 5391, Sep. 2014.
- [25] "Use of Minimal Free Energy and Self-Assembly To Form Shapes - Chemistry of Materials (ACS Publications)." [Online]. Available: <http://pubs.acs.org/doi/abs/10.1021/cm00054a028>. [Accessed: 28-Aug-2015].
- [26] 伸一中澤, 洋史比田井, and 和戸倉, "液体の界面張力を利用したプラスチックの光成形," *精密工学会誌*, vol. 68, no. 4, pp. 571–575, 2002.
- [27] J. C. Burton, F. M. Huisman, P. Alison, D. Rogerson, and P. Taborek, "Experimental and Numerical Investigation of the Equilibrium Geometry of Liquid Lenses," *Langmuir*, vol. 26, no. 19, pp. 15316–15324, Oct. 2010.
- [28] R. Aveyard, J. H. Clint, D. Nees, and V. Paunov, "Size-dependent lens angles for small oil lenses on water," *Colloids Surf. Physicochem. Eng. Asp.*, vol. 146, no. 1–3, pp. 95–111, Jan. 1999.
- [29] J. Sebilliau, "Equilibrium Thickness of Large Liquid Lenses Spreading over Another Liquid Surface," *Langmuir*, vol. 29, no. 39, pp. 12118–12128, Oct. 2013.

SURFACE SHAPE MEASURING MACHINE OF LARGE CURVED SURFACE APPLYING IMPROVED SEQUENTIAL THREE-POINT METHOD

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BACKGROUND

Demands are increasing for highly-precise shape measurement of large curved surface such as lens and reflecting mirror for telescope. The form accuracy higher than $0.1\mu\text{m}$ is required even for the parts larger than 1m. However, there is no useful method to measure precisely the shape profile of the large parts because of inevitable difficulties in preparing large tangible datum and to avoid the influence of error motions in measuring operation. Development of the measuring machine satisfying such requirements is an urgent need.

The sequential three-point method [1], which does not require the tangible datum, is a useful technique for on-machine straightness measurement of large parts. In this study, a new surface shape measuring machine to evaluate the figure profile of large curved surface is proposed using improved three-point method.

PRINCIPLE OF SURFACE SHAPE MEASUREMENT BY SEQUENTIAL THREE-POINT METHOD

Figure 1 shows the principle of sequential three-point method. The rigid sensor head, on which three displacement sensors F (front), C (center) and R (rear) are fixed at equal spaces of d , is scanned along a straight line (x -axis) on the specimen surface to be measured. The surface shape profile of the specimen is numerically calculated using the discontinuous displacement data obtained at every space of d as described in the following. The surface shape profile of the specimen and the output signals of the front, center and rear sensors are expressed by $g(x)$ and S_{3F} , S_{3C} , and S_{3R} , respectively. Supposing that the translational and rotational error motions of the sensor head are e_y and e_θ , respectively, the output signals of the sensors are expressed by Eqs. (1) to (3).

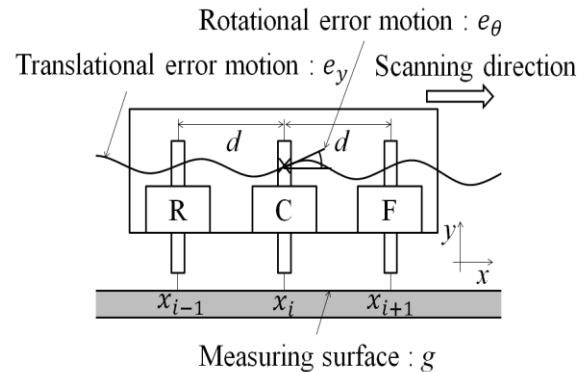


Figure.1 Principle of sequential three-point method

$$S_{3C}(x_i) = g(x_i) + e_y(x_i) \quad (1)$$

$$S_{3F}(x_i) = g(x_{i+1}) + e_y(x_i) + de_\theta(x_i) \quad (2)$$

$$S_{3R}(x_i) = g(x_{i-1}) + e_y(x_i) - de_\theta(x_i) \quad (3)$$

To eliminate the error motions e_y and e_θ , the second differential output signals is calculated as the recurrence formula Eqs. (4), where ΔS_{3F} and ΔS_{3R} are the differences between output signals of the front and center sensors and between the center and rear sensors, respectively.

$$\Delta S_{3F}(x_i) - \Delta S_{3R}(x_i) = \{g(x_{i+1}) - g(x_i)\} - \{g(x_i) - g(x_{i-1})\} \quad (4)$$

By replacing $g(x_0)$ with G_0 and difference between x_0 and x_1 with ΔG_0 , surface shape profile $g(x_n)$ without error motion is expressed by Eq. 5.

$$g(x_n) = G_0 + n\Delta G_0 + \sum_{i=1}^n (n-i) \{\Delta S_{3F}(x_i) - \Delta S_{3R}(x_i)\} \quad (5)$$

The error motions of sensor head e_y and e_θ can also be calculated from Eqs. (1) to (3) and (5). When the three sensors have different zero positions as shown in Figure 2, the zero position error α cannot be eliminated and it causes a large quadratic error component as shown in Eq. (6).

$$g(x_n) = G_0 + n\Delta G_0 + \sum_{i=1}^n (n-i) \{ \Delta S_{3F}(x_i) - \Delta S_{3R}(x_i) \} + \frac{n(n-1)}{2} \alpha \quad (6)$$

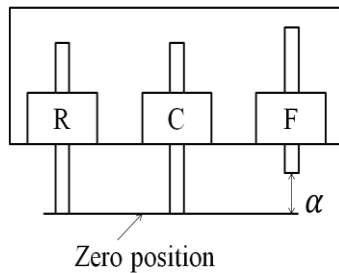


Figure.2 Zero position error between sensors

SHAPE MEASUREMENT OF CURVED SURFACE BY IMPROVED SEQUENTIAL THREE-POINT METHOD

When the measuring surface is almost flat with few undulations, the three-point method is a useful technique for on-machine measurement of its straightness because of simple and easy calculation algorithm. However, if the measuring surface has large irregularities such as convex or concave mirror, the distance between surface and sensor exceeds the working range of the sensor during measuring operation. Therefore, in improved three-point method proposed, a distance between the measuring surface and the tip of the center sensor h_a and rotational angle φ of the sensor head is adjusted so that the measuring surface always comes within the working range of all displacement sensors as shown in Figure 3. The center of rotation is set to the tip of center sensor. The output signals of the three sensors are expressed by Eqs. (7) to (9).

$$S_{3C}(x_i) = g(x_i) + e_y(x_i) - h_a(x_i) \quad (7)$$

$$S_{3F}(x_i) = g(x_{i+1}) + e_y(x_i) + de_\theta(x_i) - h_a(x_i) - d \sin \varphi(x_i) \quad (8)$$

$$S_{3R}(x_i) = g(x_{i-1}) + e_y(x_i) - de_\theta(x_i) - h_a(x_i) + d \sin \varphi(x_i) \quad (9)$$

In the same manner as Eq. (4), the second differential output signals between ΔS_{3F} and ΔS_{3R} is expressed as the recurrence formula Eq. (10).

$$\Delta S_{3F}(x_i) - \Delta S_{3R}(x_i) = \{g(x_{i+1}) - g(x_i)\} - \{g(x_i) - g(x_{i-1})\} \quad (10)$$

Eq. (10) means that both of translational movement on y -axis h_a and rotational angle φ to adjust the location and attitude of sensor head can be eliminated by the numerical calculations described above.

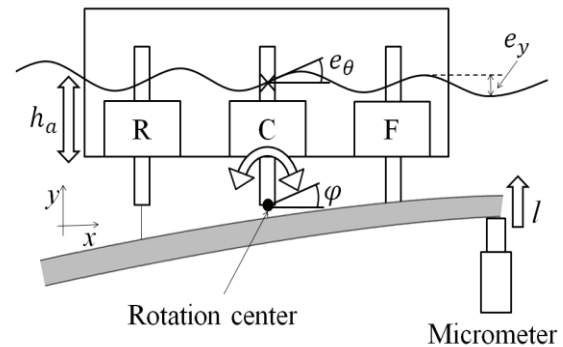


Figure.3 Principle of shape measurement of curved surface using improved three-point method

MEASURING EXPERIMENT

To ascertain the principle described above, experimental measuring equipment was assembled. The x and y axis are taken on the horizontal plane as shown in Figure 4. Three electrical comparators were used as the displacement sensors. The working range and repeatability of the sensors are $\pm 25\mu\text{m}$ and $0.3\mu\text{m}$, respectively. The distance between sensors d was 50mm. The sensor head was mounted on an adjustment unit which is a stack of a small rotary table above a fine adjustment table both of which are driven by stepping motors. The positioning resolutions of the tables were $4.0 \times 10^{-5}\text{rad}$ and $0.01\mu\text{m}$, respectively. The adjustment unit was mounted on a scanning table with a positioning accuracy of $2\mu\text{m}$ driven by a ball screw and a stepping motor.

The distance between the test piece surface and the tip of the center sensor h_a and rotational angle φ of the sensor head is adjusted by driving the stepping motors as mentioned hereunder. When the center sensor moves in the direction of x axis to a measuring position x_i from the previous position x_{i-1} , h_a is adjusted so that the output signal of the sensor approximately coincides with the central value of working range of the sensor. Subsequently, φ is adjusted so that the difference of output signals of front and rear sensors is reduced to be less than $10\mu\text{m}$.

A glass scale with smooth surface was used as a test piece to be measured. The length, width and thickness of the scale were 950mm, 22mm and 4.5mm, respectively. The scale was supported at both ends with a distance of 700mm. To produce curved surfaces, a deflection l is given at the center between the supporting points of the scale by a micrometer.

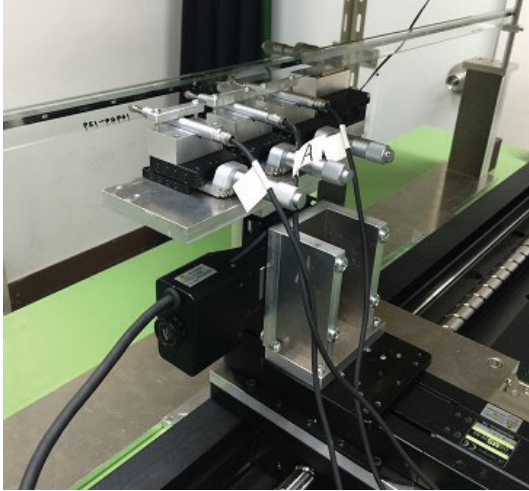


Figure.4 Measuring equipment

RESULTS OF THE MEASURING EXPERIMENT

As the accurate shape of actual test piece produced by a deflection is unknown, the measuring error of the experimental equipment is evaluated from the difference in deflection curves with different deflection amounts given onto test piece. Figure 5 show the change in theoretical deflection curves of a beam supported at both ends corresponding to different deflection amounts l , $l+\Delta l$ and $l+2\Delta l$, respectively. Beam theory suggests that the difference in deflection curves Δg_{12} and Δg_{23} are equal when the increase of deflection Δl is equal. The deflection amounts given in the experiments were 0.2mm, 0.7mm and 1.2mm and the increase was 0.5mm, respectively.

Figure 6 shows the shape profiles of the test piece surface measured by the method proposed. $g_1(x)$, $g_2(x)$ and $g_3(x)$ are the shape profiles corresponding to the deflections of $l_1 = l = 0.2\text{mm}$, $l_2 = l+\Delta l = 0.7\text{mm}$ and $l_3 = l+2\Delta l = 1.2\text{mm}$, respectively. As the increases in deflection amount are equal, the difference between Δg_{12} and Δg_{23} should be zero over the whole measuring length. If Δg_{12} and Δg_{23} are not equal, the difference between Δg_{12} and Δg_{23} is an indicator of measuring error of the measuring method proposed.

Figure 7 shows the difference between Δg_{12} and Δg_{23} . The difference seems to include quadratic error component due to zero position error of the sensors. Therefore, the difference after eliminating the approximate value of zero point error is shown Figure 8. The width of fluctuation of about $0.45\mu\text{m}$ observed from three trials is considered to be the measuring error of the equipment.

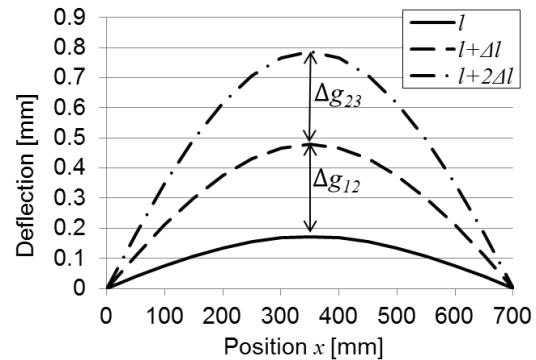


Figure.5 Change in theoretical deflection curves of a beam supported at both ends

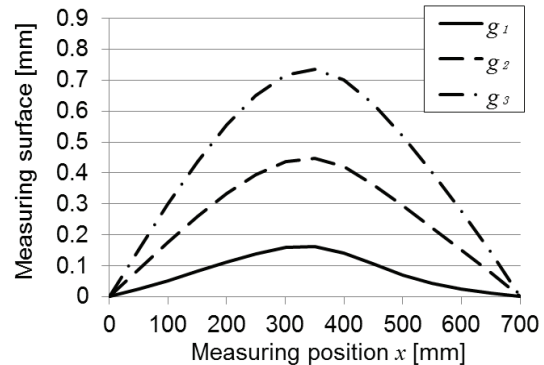


Figure.6 Shape profiles of the test piece with different deflection amounts

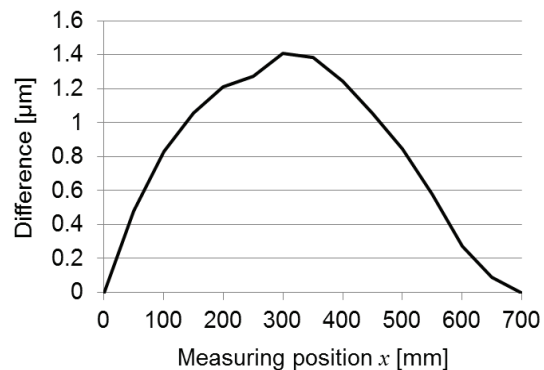


Figure.7 Difference between Δg_{12} and Δg_{23} of the test piece

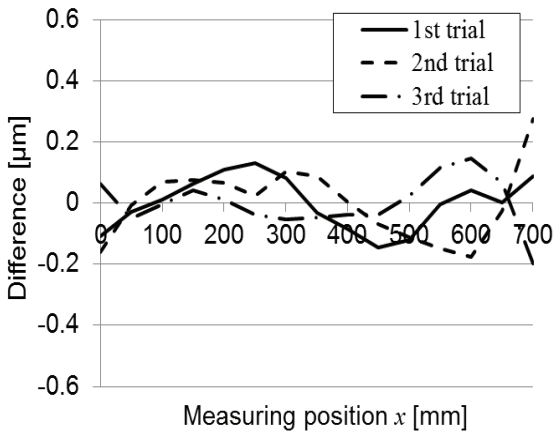


Figure.8 Estimated measuring error of the equipment

INFLUENCE OF ERROR MOTION OF SCANNING TABLE AND ROTATION OF SENSORE HEAD ON MEASUREMENT ERROR

Another superior feature of the method is that the error motion of table does not deteriorate the shape profile. Figure 9 shows the error motion of scanning table measured by a displacement sensor using a straight edge as a datum. A periodic error motion the amplitude of which is about 6 μm is observed corresponding to the pitch of ball screw using in driving table. However, no deterioration of shape profile due to the error motion was observed in Figure 8.

Figure 10 shows a shift of measuring position due to rotation of sensor head. A geometrical analyses suggests that the measuring error due to the position shift is estimated to be less than $0.025\mu\text{m}$ even when the deflection amount is 1.2mm as shown in Figure 11.

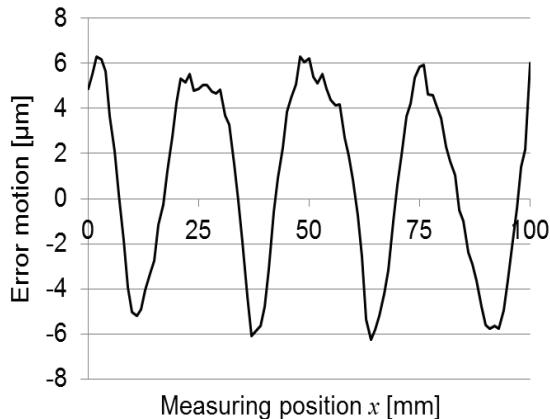


Figure.9 Error motion of scanning table

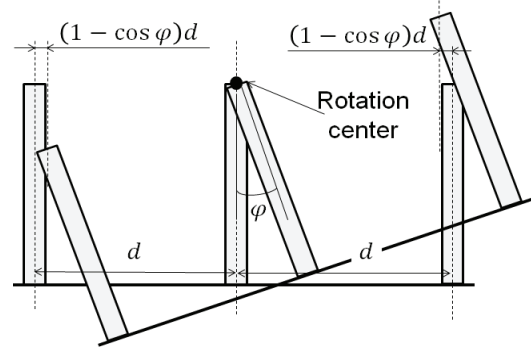


Figure.10 Shift of measuring position due to rotation of sensor head

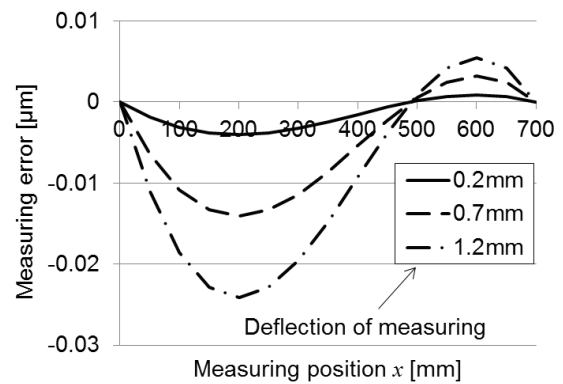


Figure.11 Influence of position shift on measuring error

SUMMARY

A design concept of new surface shape measuring machine to evaluate the shape profile of large curved surface is proposed using improved three-point method. The experimental measuring equipment assembled based on the concept suggests that the method has enough measuring accuracy even by using scanning table with considerably large error motion. To improve the influence of zero point error of the sensors is the subject at present.

REFERENCES

- [1] Obi, M., Furukawa, S. : A study of the measuring for straightness using a sequential point method", 1991, Trans. JSME, C-57, 542, 3197-3201.
- [2] Tamagawa, T., Uda, Y., Shimada, S., Imura, R. : Development of surface shape measuring machine of large optics by sequential three-point method, 2014, Proc. Japan Society for Precision Engineering 2014 Autumn Meeting, 823-824. (in Japanese)

PRECISE NON-CONTACT CENTER THICKNESS METROLOGY

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ABSTRACT

The last place an optics manufacturer wants to physically touch a lens is at the center. However, this is precisely what is currently done to measure center thickness of lenses. Using contact methods, the question is not whether the optic is damaged, it is whether the resulting damage is acceptably low. At Bridger Photonics, we have proven the feasibility of a non-contact center thickness metrology system to address this need. The apparatus uses a technique similar to swept-frequency optical coherence tomography to measure both physical thickness and optical thickness. From these measurements, the group index of refraction can also be determined. Moreover, the phase index can be determined, given the Sellmeier coefficients. In this presentation, we will report our demonstrated measurement range of 75 mm optical thickness (larger possible), as low as 20 nm precision, and group index of refraction determined to better than 5 parts in 10^5 . We believe the metrology system resulting from these proof-of-principle demonstrations will be a valuable tool for precision optics manufacturing.

INTRODUCTION

In optics manufacturing, scrapped or reworked parts can often represent significant expenditures that inflate cost of production, particularly for custom optics manufacturing and prototyping. While non-contact methods of optical surface metrology exist to characterize, for instance the surface figure and irregularity of lenses, no such analogous methods exist for characterizing the lens center thickness, except for thin and flat optics. Instead, optics manufacturers typically rely on mechanical means to measure the center thickness such as micrometers and drop gauges [1]. Unfortunately, physical contact at or near the center of the lens inevitably results in damage within the clear aperture. The only question is whether the level of damage is within the tolerance of the manufacturing specifications (often determined by the customer). Contact at the lens center is particularly worrisome for high-power laser

applications where imperfections in the lens can seed damage propagation and act as damage centers [2].

Moreover, while center thickness uncertainty tolerances are typically 25 μm or greater, the most demanding optical designs now require 1 micron or better center thickness uncertainty. This means the metrology of the center thickness should offer about 200 nm uncertainty or better to provide adequate margin for error. While a number of optical techniques such as chromatic confocal and low coherence interferometry can measure surfaces with this required high precision and accuracy, these techniques struggle to accurately measure over thickness ranges much beyond several millimeters[3]. Furthermore, it is typically the physical thickness rather than the optical thickness that is specified and is most desired in center thickness measurements. And while it is possible to counter-propagate two of these measurement systems to independently measure the location of both sides of an optic, this method puts an extreme premium on the mechanical and environmental stability of the separation between the two measurement probes.

Here we present a frequency-modulated continuous-wave (FMCW) laser detection and ranging measurement technique to overcome these challenges and offer a non-contact center thickness metrology system with high precision and accuracy.⁴ Our configuration is self-calibrating and has the potential to measure all optical thicknesses that we have found practical for lens manufacturing (up to 75 mm thicknesses demonstrated, but larger possible). Moreover, the technique enables measurement of both the physical thickness and the group index of the material, the latter of which we have found valuable for manufacturers to detect incorrect materials before or during processing.

FMCW LADAR BACKGROUND

The FMCW Ladar technique is summarized in Figure 1. For this technique, a laser is chirped (frequency swept) over a given bandwidth, B . Light from the chirped laser is split into two portions. One portion

of the chirped laser light is transmitted (Tx) to the optic under test, while the other portion serves as a local oscillator (LO). The light returning (Rx) from the optic under test is collected and interferometrically combined with the local oscillator. The combined beam is directed onto a photodetector and an RF beat note results from the time delay between the LO and Rx at a frequency given by $f_{beat} = K\tau$, as shown in Figure 1 (bottom, left), where K is the chirp rate (often in MHz/μs), and τ is the time delay between the LO and Rx. With accurate knowledge of the chirp rate and measurement of the beat note frequency, f_{beat} , the time delay, and therefore the distance, to each surface of the optic can be determined. From the time delay, the range to target is given by $R = \tau c / 2$, where c is the speed of light. A Fourier Transform of the detected time-domain signal yields the range profile shown at the bottom right of Figure 1.

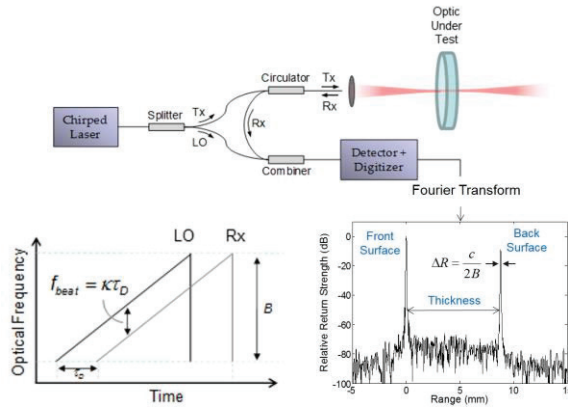


Figure 1. Top: FMCW ladar technique for measuring thickness of optics. Bottom left: Time-domain picture of the optical frequencies of the LO and Rx waveforms as functions of time. Bottom right: Frequency-domain range profile showing a high-fidelity peak for each surface. The width of the peaks is determined by the chirp bandwidth.

The range resolution of the measurement, defined as the minimum resolvable range separation (full width at half maximum) between two targets on the same bearing, is given by $\Delta R = c / 2B$, where B is the total bandwidth of the chirp. This equation makes clear the desire for larger bandwidth in order to achieve better resolution. However, it is only valid over measurement windows that are well within the laser coherence length. It is also only valid when the optical frequency chirp is sufficiently linear. Bridger uses active linearization of the optical frequency chirps to achieve resolution-limited peak shapes. The range precision (also known as the repeatability or reproducibility) depends on the signal-to-noise ratio

(SNR) and is defined as the standard deviation of a statistically significant number of range measurements of the same target under the same conditions. Mathematically, a lower limit for the range precision is given by the Cramér-Rao relation $\sigma \approx \Delta R / \sqrt{SNR}$. The measurement precision that a system can achieve can therefore be far better than its resolution. Moreover, by averaging over multiple measurements the precision can be further improved as $1/\sqrt{N}$, where N is the number of measurements, as long as the noise processes are stochastic and uncorrelated. Finally, due to the high coherence and active linearization of the chirped lasers, measurements can be performed on virtually all thicknesses that are practical for optics manufacturing ($\gg 10$ cm if needed and configured properly).

EXPERIMENT AND RESULTS

For baseline testing, Bridger used the setup shown in the schematic of Figure 1 (top) and in the image in Figure 2 (left). The Tx light is focused onto a nominally 12 mm thick uncoated UV fused silica window with parallel faces. The confocal parameter of the focused beam was roughly selected to match the window thickness and the window was aligned using a kinematic mount to reflect Rx light directly back into the transmitted optical mode. The chirp bandwidth we used for the present experiments was 150 GHz, which yields 1 mm range peak widths, but we have demonstrated up to 6 THz (at slower rates) in other experiments. The system repeatedly produced chirps at an update rate of 4 kHz. Using this setup, we generated range profiles featuring one peak for each of the optical surfaces, as in Figure 1 (bottom right). We fit the peaks to Gaussian functions and determined the centers of the two fit peaks. We obtained the group optical thickness of the sample by taking the difference between the two peak fit peak centers. The precision of these measurements was 340 nm as shown in Figure 2 (left). Next, we averaged over successive measurements to produce the Allan deviation plot shown in Figure 2 (right). The plot shows the expected $1/\sqrt{N}$ behavior down to about 200 measurements, after which environmental and mechanical noise sources are assumed to begin influencing the behavior because no extraordinary measures were taken to isolate the experiment. Nevertheless, for a measurement time of about 50 ms, the system reaches 20 nm precision. This baseline experiment showed that the system could achieve acceptable precisions in short measurement times. The accuracy of the

measurement is determined by the chirp rate of our laser and is typically calibrated to a few parts in 10^6 , but can be calibrated better if needed. The downside to using only one transceiver is that this baseline testing configuration only provided group optical thickness and not physical thickness.

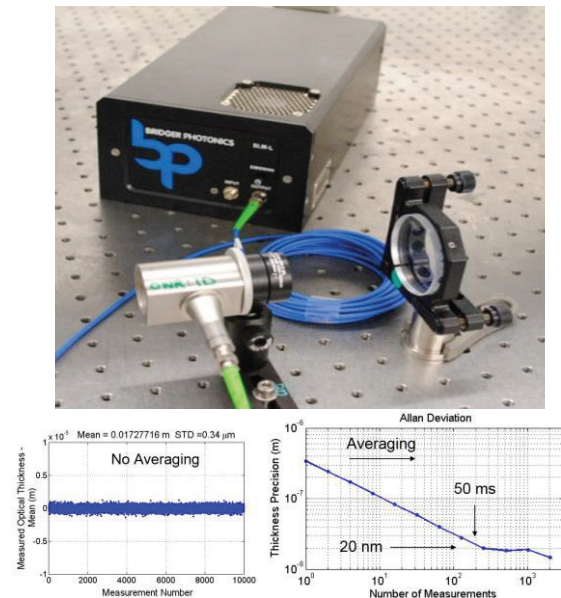


Figure 2. Top: Image of setup for baseline testing. Left: Single-shot precision of 340 nm for a UV fused silica optic with nominal physical thickness of 12 mm. Right: Allan deviation showing that 20 nm precision can be achieved after about 50 ms of averaging.

To enable physical thickness measurements we used two counter-propagating beams emitted from opposing transceivers, as shown in the image of Figure 3 (top) and the simplified schematic shown in Figure 3 (left). Each of the two measurement beams was aligned into the opposing transceiver to ensure spatial overlap of the two beams. The confocal parameters were set to 75 mm to enable measurement of thicker optics, if desired. The measurement procedure was then as follows. First, the sample was removed and at least one of the two measurement beams was used to measure the separation between two reference surfaces; shown as “A” in Figure 3 (left). Second, each measurement beam measured the distance from its nearest reference surface to the nearest surface of the optic under test. These separations are shown as “B” and “C” in Figure 3 (left). The physical thickness is then obtained by subtracting B and C from A.

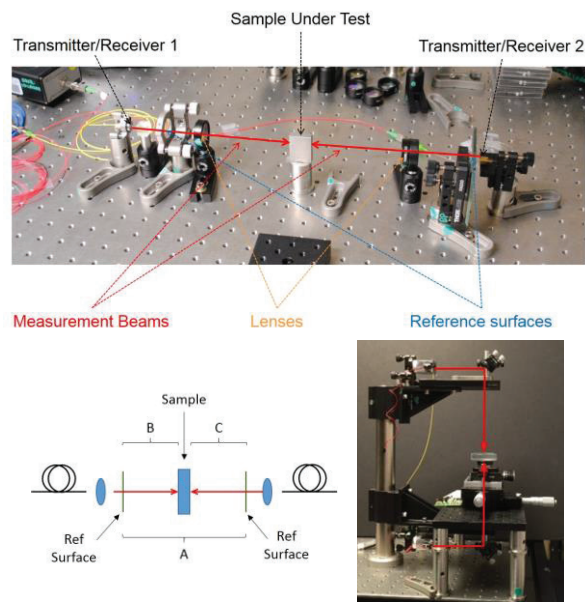


Figure 3. Top: Counter-propagating beam setup to measure physical thickness. Left: Simplified schematic of measurement procedure. Right: Vertical beam configuration.

The measurement of the reference surface spacing is an important self-calibration step that ensures robustness of the technique. Without this measurement, any drift between the two transceivers, due to mechanical or environmental perturbations, for instance, will cause measurement errors in the location of each sample surface. By measuring the reference surface separation either shortly before or simultaneously with the sample measurement, drift or other perturbations to the transceivers is accounted for. Moreover, by measuring both the front and back surfaces of the sample optic with at least one measurement beam, the group optical thickness can also be obtained. Finally, with the physical thickness and the group optical thickness, the group refractive index of the material can be calculated. Figure 3 (right) shows a subsequent setup with vertical beam alignment so the optic did not require rigid mounting. This vertical configuration uses the delivery fiber distal ends as the reference surfaces to further simplify the setup.

Using the setup shown in Figure 3 (top), we measured the thickness of several different sample optics, the results of which are shown in Figure 4. For this setup, the range peaks associated with the additional surfaces that enable the physical thickness measurement results in a degradation of the measurement precision. Also, for these measurements, our data acquisition was slowed by

the available hardware for this experiment to about 1% of the possible data acquisition rate. Figure 4 (top) shows the same uncoated UV fused silica optic whose group optical thickness was measured previously. For this optic, the precision of each surface reached between 100 nm and 200 nm after 200 averages. The group optical thickness was determined with a precision of 150 nm and the physical thickness was determined with a precision of 180 nm. Using the group optical and physical thickness measurements, the group refractive index was calculated 1.4626 ± 0.0001 for the operating wavelength of 1550 nm. Figure 4 also shows the analogous measurements for an uncoated concave/convex lens with 500 averages (center), and an anti-reflection (AR) coated optic with a ground glass back surface for 5,000 averages (bottom).

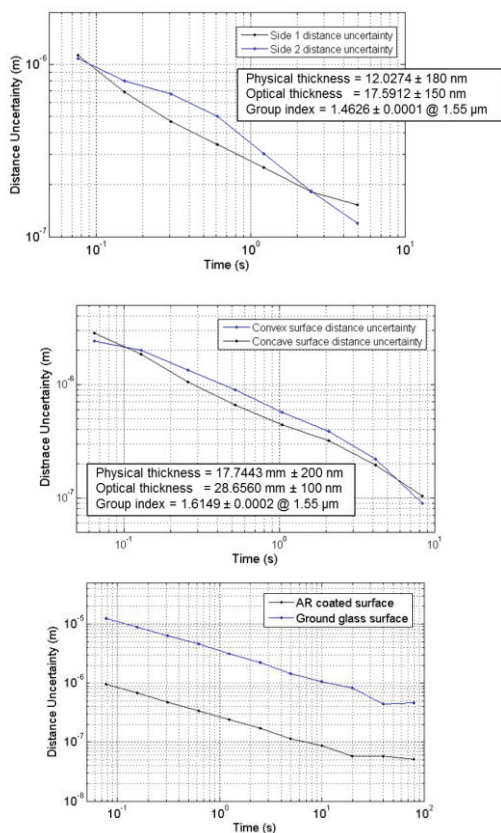


Figure 4. Allan deviation measurements for the two surfaces of three different optics. **Top:** Uncoated UV fused silica window with 12 mm nominal thickness. **Center:** Uncoated aspheric lens with one concave and one convex surface. **Bottom:** dielectric mirror with one anti-reflection coated surface and one ground glass surface.

CONCLUSIONS

We have demonstrated a self-calibrating thickness metrology system based on FMCW lidar that is capable of measuring large thicknesses for samples with flat or curved surfaces applicable to optics manufacturing. The system uses measurements from two counter-propagating transceivers. The demonstrated self-calibration removes the measurement sensitivity to drift and other changes in the separation of the transceivers. We achieved precisions of 200 nm or better for the polished optical surfaces that we measured. We also showed that the system is capable of measuring samples with rough surfaces.

REFERENCES

- [1] see, for instance, <http://ecatalog.mitutoyo.com/Height-Gages-C1262.aspx>
- [2] A. H. Guenther, D. Milam, and B. E. Newnam, "Laser Induced Damage in Optical Materials," Proceedings of a Symposium Sponsored by the American Society for Testing and Materials and by the National Bureau of Standards, Edited by H. E. Bennett, 1985.
- [3] F. Blateyron, Chromatic Confocal Microscopy, in Optical Measurement of Surface Topography, (Springer Berlin Heidelberg) pp 71-106 (2011)
- [4] Stone W, Juberts M, Dagalakakis N, Stone J, Gorman J. Performance Analysis of Next Generation LADAR for Manufacturing, Construction and Mobility. National Institute of Standards and Technology, NISTIR 7117, 2004.

UNDERSTANDING ARTICULATING ARM LASER LINE SCANNERS FOR PRECISION ENGINEERING: OPERATOR USAGE EFFECTS

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ABSTRACT

This paper reports initial results from the National Physical Laboratory's 3D optical scanner assessment facility, where the effect the operator may have on the performance of articulated arm mounted 3D laser line scanners has been investigated. Two scanners from different manufacturers have been assessed. Reliable and repeatable results are produced by mounting each system on an automated, motorised, adjustable rail system, allowing for the control of scan: direction, velocity and distance to target. These results indicate that both scanners assessed perform within the manufacturers' specifications with varying scanner head velocity but, for the best results with these scanners, the scanner head should not be moved with a linear velocity greater than $16 \text{ mm} \cdot \text{s}^{-1}$.

INTRODUCTION

Cartesian co-ordinate measuring machines (CMMs) are well established metrology instruments. [1] They are used far and wide for the measurement and inspection of an enormous range of parts, with some instruments claiming accuracies in the sub-micrometre range. [2]

Articulated arm mounted 3D laser line scanners offer a relatively new and developing method of measuring objects. Currently, articulated arm mounted 3D laser line scanners, often referred to as 3D laser line scanners, do not offer sub-micrometre level accuracy. Unfortunately, these scanners lack the metrological infrastructure to provide measurement traceability back to the SI metre, no ISO standards are available and measurement uncertainties cannot be calculated. However, 3D laser line scanners are popular tools for carrying out measurements of parts.

The National FreeForm Centre at the National Physical Laboratory (NPL, UK) recognises the need for traceability of 3D optical scanners in

industry and has developed an optical scanner assessment facility. [3] The facility is designed to provide, to scanner manufacturers or users, an independent assessment of the performance of a particular 3D optical scanner. This is also a longer term aim to promote best practice and attempt to provide traceability by understanding the uncertainly components that affect 3D optical scanner dimensional measurements.

Previous work has shown that 3D laser line scanners can be affected by a wide range of influences which can be broadly split into two categories; "operator effects" and "environment effects". Environment effects can include ambient light conditions [4], the physical size of features on the test surface and optical properties of the object being measured. [5] This paper discusses those categorised as operator effects and reports the effect of scanner head velocity on measurement repeatability.

OPERATOR EFFECTS

3D laser line scanners operate by projecting a laser line onto the surface to be measured. An off axis camera is focused at a particular point where the laser line is expected to reflect off the surface. When pointed at a flat surface, the laser line appears flat to the camera but when pointed towards a curved surface the line appears deformed. The camera measures this deformation and calculates the shape of the underlying surface that causes this deviation from the projected straight line. By moving this line along a surface it is possible to measure the surface profile and from this create a 3D model of the physical part.

3D laser line scanners mounted on articulated arms are designed to be manually handled, relying on a user to move the laser line along the surface to be measured. This human interaction can and does have an effect on the scanner's output. Each 3D laser line scanner has an associated scanner refresh rate *i.e.* the frequency with which it can take measurements.

The speed with which the scanner is moved across the surface will affect the amount of useful data the scanner can acquire. Each scanner acquires data at a fixed acquisition rate. When moved quickly across the surface of the object being scanned, the density of the acquired data is low. When moved more slowly, the density of the acquired data is higher. The ideal situation is where the scanner is moved sufficiently slowly to acquire sufficient data while not moving too slow which will increase inspection time and hence cost.

Additionally, 3D laser line scanners have a field of view over which the camera records the shape of the projected laser line on the object. By moving the scanner towards and away from the surface being measured, the operator can change where, within an allowed “field of view” the scanner is recording data from. As optical lenses have an optimal path length, any deviation from the optimum position may have an effect on the quality of data acquired. This paper reports a system developed by NPL to find the optimum balance between scan velocity and measurement quality.

LASER SCANNERS TESTED

The dimensional performance of two, similar articulated arm mounted 3D laser line scanners was tested: a FARO FaroArm Platinum with ScanArm v3 laser probe and a Hexagon Romer arm with ScanWorks v5 Perceptron laser scanner. For the purposes of impartiality, the systems are referred to as “system A” and “system B”, in no particular order.

3D LASER LINE SCANNER TESTING SYSTEM

To allow for repeatability of measurements and to remove operator influence on the results, the human user is removed from the measurement procedure. This was achieved by the creation of an automated scanning system which consisted of an adjustable frame with a precision carriage. The 3D laser line scanner was fixed in place on an optical table and the scanning head attached to the automated scanning system. A NEMA 17 stepper motor, controlled using a Raspberry Pi Model B+ microcontroller, maintains a constant scanning velocity. The ball bar test object was manufactured from two 48 mm diameter spheres mounted on 20 mm carbon fibre box section, with a separation of ~330 mm, with an additional alignment sphere placed between the two larger balls, which can be positioned at various heights

from the scanner head and varying angles to the plane of the scanner motion. *FIGURE 1* is a photograph showing the automated scanning system.

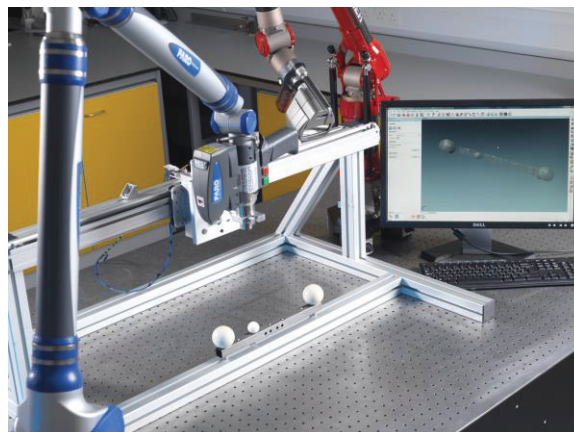


FIGURE 1. Photograph showing the automated scanning system as well as the two scanners which were tested.

Both 3D laser line scanners were connected to a desktop PC running Polyworks 2014. A CAD model of the scanned test object was imported and nominal spheres defined. Point clouds were collected by moving the 3D laser line scanner over the surface. This data was then aligned to the CAD model and information about the size and separation of the spheres extracted. Among the recorded information were the diameters of the two 48 mm spheres, their separation and the coordinates of their centres in x, y and z.

Approximate scanner head velocities were determined through calculations using the stepper motor pulse settings and then cross checked using a stopwatch. The motor is capable of producing a maximum velocity of $\sim 90 \text{ mm}\cdot\text{s}^{-1}$, using a 1 ms delay between the steps of the motor. This velocity can then be decreased by increasing this delay. By doubling this delay the velocity was decreased to a minimum of $\sim 0.125 \text{ mm}\cdot\text{s}^{-1}$.

SCAN VELOCITY RESULTS

The distance between the two 48 mm balls was calculated at the minimum scanner head velocity, $\sim 0.125 \text{ mm}\cdot\text{s}^{-1}$, and all the subsequent distances were compared to this and the difference between the measurements taken.

The results displaying the effect of scanner head velocity has on the dimensional measurements produced are presented below, with System A's

results show in *FIGURE 2* and System B's results shown in *FIGURE 3*. Note the larger

scale in *FIGURE 3*.

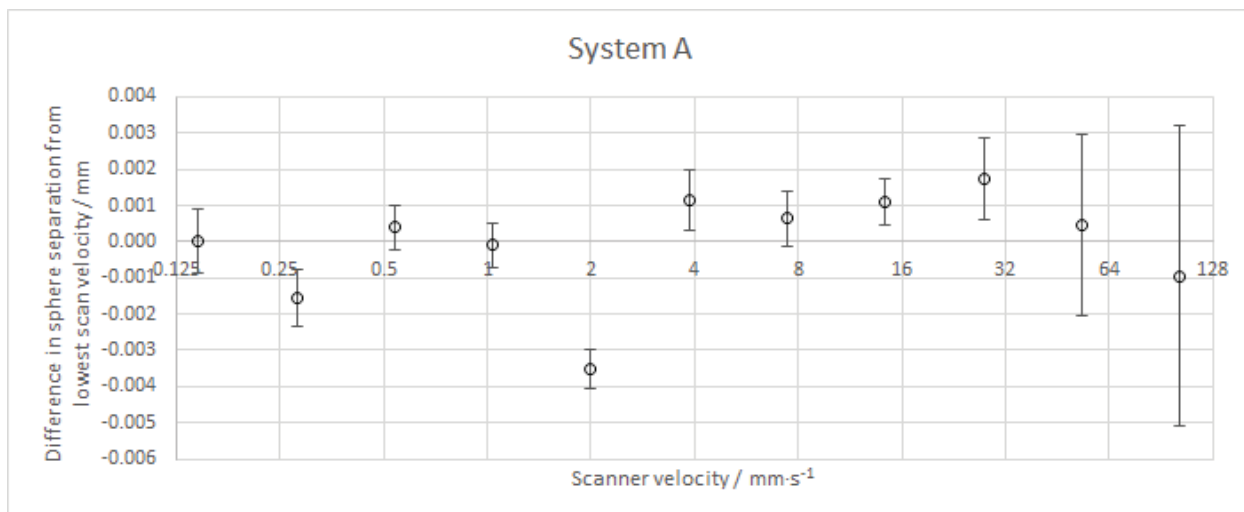


FIGURE 2. Graph showing averaged values of sphere separation distances for System A. Error bars display the standard deviation of the 20 results collected at each velocity.

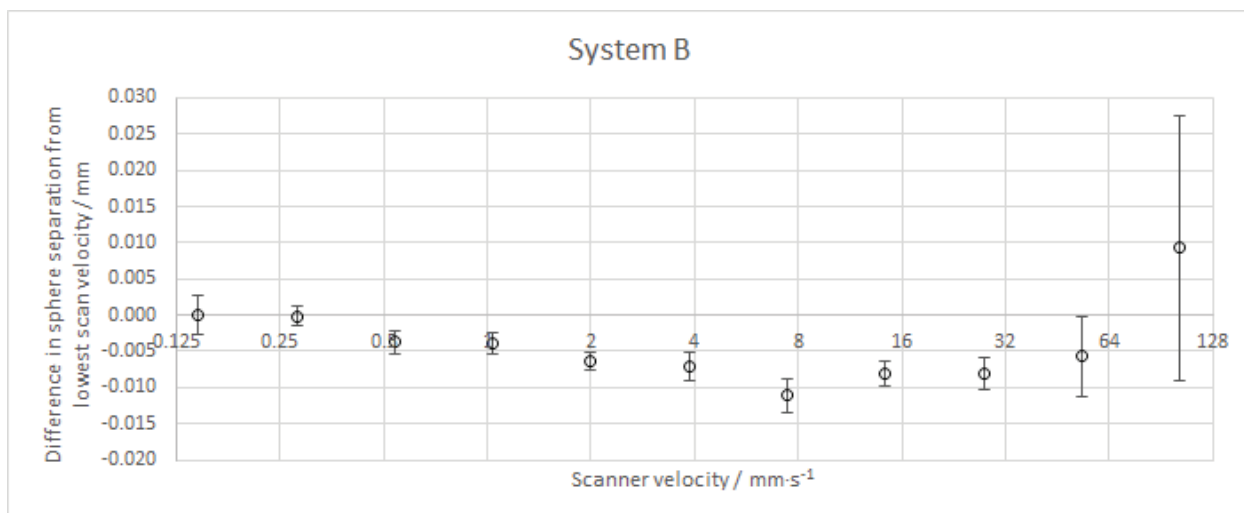


FIGURE 3. Graph showing averaged values of sphere separation distances for System B. Error bars display the standard deviation of the 20 results collected at each velocity.

FIGURE 2 shows that difference in the sphere separation of the spheres, when measured using System A, differs by a maximum of just less than 6 μm . The variation of the sphere separation distance is relatively constant under changing scan velocity. It can be seen, however, that the standard deviation of the measured sphere separations does appear to increase with scan velocity. At velocities less than 16 $\text{mm}\cdot\text{s}^{-1}$ the standard deviation of the various measurements is relatively constant. However, once the scanner head velocity rises above

16 $\text{mm}\cdot\text{s}^{-1}$, the standard deviations can be seen to increase.

FIGURE 3 shows that difference in the sphere separation, when measured using System B, differs from the reference by a maximum of less than $\sim 12 \mu\text{m}$. Similarly, an increase in standard deviation when the scanner head is moved at a velocity of greater than 32 $\text{mm}\cdot\text{s}^{-1}$ is seen.

CONCLUSIONS

Both System A and System B appear to display relatively stable results under varying scanner head velocities. This indicates that when both systems are used as intended, in the hand of an operator, reliable results will be able to be captured.

According to literature released by the manufactures; the Faro ScanArm v3 laser probe should achieve an “accuracy” of $\pm 35 \mu\text{m}$ [6] and the ScanWorks v5 Perceptron laser scanner should achieve a “measurement accuracy” of $24 \mu\text{m}$. [7] As discussed previously, there are no formal standards for optical CMMs to conform to, so the words “accuracy” and “measurement accuracy” don’t have definitions here. However, the results collected show that System A and System B both achieve results well within these quoted values.

It should be noted, however, that while the reported results are well within the quoted “accuracies” for both systems, they are the mean values of repeated measurements. Both System A and System B displayed increased standard deviations at velocities above $16 \text{ mm}\cdot\text{s}^{-1}$, with System B displaying a standard deviation of almost $30 \mu\text{m}$ when the scanner head was moved at $\sim 90 \text{ mm}\cdot\text{s}^{-1}$. This result is significant as most laser line scanners are used as part of the quality control procedure, where only one measurement will be taken. This large standard deviation could lead to parts being needlessly rejected or incorrectly accepted if the scanning velocity is too high. With this in mind it is important that when these pieces of equipment are used as part of a quality control procedure, a sufficiently slow scan velocity is used to ensure sufficiently accurate data is collected.

Analysis of the point cloud data sets returned by the scanners shows that the measured points comprise a series of parallel lines across the surface of the ball bar. As the scanning head velocity is increased the separation between these lines increases. It is the refresh rate of the laser scanning head that causes this change in line separation. Newer laser scanning systems have much faster refresh rates and as such may produce more repeatable results at higher scan head velocities. Further investigation is required to confirm this supposition.

Laser scanner systems often deploy some form of feedback system to assist the operator in maintaining an optimum offset between the scanner head and the object being scanned. Laser scanner systems have a small range of distances from the scanned surface over which they can operate. These scanner head velocity tests were carried out with the spheres of the ball bar positioned in the middle of this field of view. The effect the position of the object within the field of view is currently being assessed and will be reported in a later paper.

REFERENCES

- [1] Flack D, 2011 *CMM verification* (National Physical Laboratory) Good Practice Guide No 42
<http://www.npl.co.uk/publications/cmm-verification>
- [2] ISO 10360-2:2009 *Geometrical product specifications (GPS) -- Acceptance and reverification tests for coordinate measuring machines (CMM) -- Part 2: CMMs used for measuring linear dimensions*
- [3] Dury M, Brown S, McCarthy M, Woodward S. *3D Optical Scanner Dimensional Verification Facility at the NPL's "National FreeForm Centre"*. Submitted to Lambdamap, University of Huddersfield, March 2015
- [4] Lemeš S, Zaimović-Uzunović N. Study of ambient light influence on laser 3D scanning. *7th International Conference on Industrial Tools and Material Processing Technologies*; 4 - 7 October 2009; Ljubljana, Slovenia. pp. 327-30
- [5] Zaimović-Uzunović N, Lemeš S. Influences of surface parameters on laser 3D scanning. *10th International Symposium on Measurement and Quality Control*; 5 - 9 September 2010; Osaka, Japan. pp. 408-11
- [6] *Measuring Arm FARO ScanArm – Portable Measurement Solutions from FARO*
<http://www.faro.com/en-us/products/metrology/measuring-arm-faro-scanarm/overview>
- [7] *Portable 3D Scanning – Perceptron Inc., ScanWorks by perceptron*
<http://perceptron.com/products/3d-scanning-solutions/portable/>

SELF-CALIBRATION OF THE NON-LINEAR LENGTH DEVIATION OF A LINEAR ENCODER BY USING TWO READING HEADS

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INTRODUCTION

To measure linear motion in high-accuracy machine tools, CMMs and other inspection machines, linear encoders and laser interferometers are mostly utilized. Linear encoders, however, are the preferred choice because of their lower cost and easier integration with respect to laser interferometers, as well as their lower sensitivity to environmental changes. Nevertheless, to reach sub-micrometer or even nanometer accuracies, calibration of the linear encoders is indispensable. Calibration is mostly done by comparison with a laser interferometer [1], but if one is only interested in the non-linear length deviation of the scale instead of the absolute length deviation, self-calibration might be an easier and more cost-effective approach. Self-calibration does not require an external length transfer standard to derive the length deviation. Although this calibration procedure has already been investigated for angle encoders [2, 3, 4], little or no research can be found on the self-calibration of linear encoders.

This paper proposes a method to perform self-calibration of a linear encoder to determine the deviations from linearity by using two reading heads. The calibration concept will be explained and a numerical simulation will give more insight in the calibration uncertainty. Experiments on a 170-mm Heidenhain LIP281 encoder will show the reproducibility of the proposed method.

SELF-CALIBRATION METHOD

The deviation of the measured value of the encoder from the true value is constituted by a *scale error*, which is the linear deviation from the true value, and by a *non-linear deviation*, as depicted in Figure 1. The calibration method discussed in this paper focuses on the calibration of the non-linearity.

The self-calibration procedure, illustrated by Figure 2, is performed as follows. First, two reading heads that are located a distance D_{12} apart are translated with respect to the scale. The

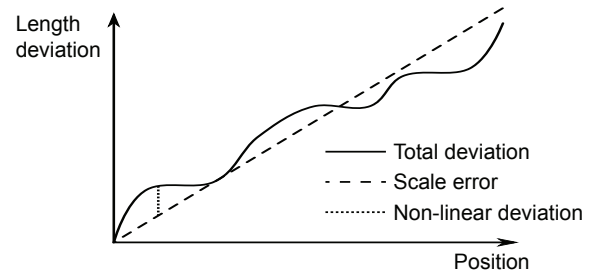


FIGURE 1. Definition of the length deviation of displacement measurements.

displacement measurement $x_{i,1}$ of each reading head $i \in 1, 2$ can be expressed as:

$$x_{1,1}(x_1^*) = x_1^* - e(x_1^*) - a \cdot x_1^* - b \quad (1)$$

$$x_{2,1}(x_1^* + D_{12}) = x_1^* + D_{12} - e(x_1^* + D_{12}) - a \cdot x_1^* - b \quad (2)$$

with x_1^* the true displacement of the scale, $e(x)$ the non-linear length deviation of the scale in function of the position x on the scale, a the scale error and b a constant deviation. Next, one of the two reading heads is displaced over a distance D_{23} and the measurement is repeated, leading to the following equations:

$$x_{1,2}(x_2^*) = x_2^* - e(x_2^*) - a \cdot x_2^* - b \quad (3)$$

$$x_{2,2}(x_2^* + D_{12} + D_{23}) = x_2^* + D_{12} + D_{23} - e(x_2^* + D_{12} + D_{23}) - a \cdot x_2^* - b \quad (4)$$

Let x_j^* be discretized at intervals Δx . It is assumed that $e(x_j^*) \approx e(k \cdot \Delta x)$ and $x_{i,j}(x_j^*) \approx x_{i,j}(k \cdot \Delta x) = x_{i,j}(k)$, with $k \in \mathbb{N}$. This assumption is viable since the length deviation profile is sufficiently smooth. Then, from these equations, x_1^* and x_2^* can be eliminated, giving:

$$e(k_1) - e(k_1 + d_{12}) + D_{12}(1 - a) = x_{2,1}(k_1) - x_{1,1}(k_1) \quad (5)$$

$$e(k_2) - e(k_2 + d_{12} + d_{23}) + \dots (D_{12} + D_{23})(1 - a) = x_{2,2}(k_2) - x_{1,2}(k_2) \quad (6)$$

$$k_1 \in [1, l - d_{12}], k_2 \in [1, l - d_{12} - d_{23}]$$

with $D_{12}/\Delta x = d_{12}$, $D_{23}/\Delta x = d_{23}$ and $L/\Delta x = l$, L being the measurement length of the scale. Since scale error is not considered, the linear trend of $e(k)$ should be removed. Moreover, the mean value of $e(k)$ is set to zero, resulting in:

$$\sum_{k=1}^{l+1} k \cdot e(k) = 0 \quad (7)$$

$$\sum_{k=1}^{l+1} e(k) = 0 \quad (8)$$

Equations (5) to (8) can then be written as:

$$\begin{bmatrix} \mathbf{P} \\ \mathbf{A}_1 & \mathbf{1} & \mathbf{0} \\ \mathbf{A}_2 & \mathbf{0} & \mathbf{1} \\ \mathbf{W} & \mathbf{0} & \mathbf{0} \end{bmatrix} \begin{bmatrix} \mathbf{y} \\ \mathbf{e} \\ C_1 \\ C_2 \end{bmatrix} = \begin{bmatrix} \mathbf{m} \\ \mathbf{x}_{2,1}(\mathbf{k}_1) - \mathbf{x}_{1,1}(\mathbf{k}_1) \\ \mathbf{x}_{2,2}(\mathbf{k}_2) - \mathbf{x}_{1,2}(\mathbf{k}_2) \\ \mathbf{0} \end{bmatrix} \quad (9)$$

with

$$\mathbf{A}_1 = \begin{bmatrix} \mathbf{I}_{l-d_{12}+1} & \mathbf{0}_{(l-d_{12}+1) \times d_{12}} \\ \mathbf{0}_{(l-d_{12}+1) \times d_{12}} & \mathbf{I}_{l-d_{12}+1} \end{bmatrix} \quad (10)$$

$$\mathbf{A}_2 = \begin{bmatrix} \mathbf{I}_{l-d_{13}+1} & \mathbf{0}_{(l-d_{13}+1) \times d_{13}} \\ \mathbf{0}_{(l-d_{13}+1) \times d_{13}} & \mathbf{I}_{l-d_{13}+1} \end{bmatrix} \quad (11)$$

$$\mathbf{W} = \begin{bmatrix} 1 & 2 & \cdots & l & l+1 \\ 1 & 1 & \cdots & 1 & 1 \end{bmatrix} \quad (12)$$

$$C_1 = D_{12}(1 - a) \quad (13)$$

$$C_2 = (D_{12} + D_{23})(1 - a) \quad (14)$$

The system of linear equations (9) is over-determined and can be solved in a least-squares sense by

$$\mathbf{y} = (\mathbf{P}^T \mathbf{P})^{-1} \mathbf{P}^T \mathbf{m} \quad (15)$$

The non-linear length deviation of the scale is then equal to

$$\mathbf{e} = [\mathbf{y}(1) \quad \mathbf{y}(2) \quad \cdots \quad \mathbf{y}(l+1)] \quad (16)$$

In order for this least-squares problem to have a solution, D_{12} and D_{23} should be chosen such that the greatest common divisor of d_{12} and $d_{12} + d_{23}$ equals 1. Moreover, a larger D_{23} decreases the sensitivity of the least-squares estimate to measurement errors in $x_{i,j}(k)$.

NUMERICAL SIMULATION

The uncertainty of the solution of the least-squares estimate can be derived from

$$u_{\mathbf{e}(k)} = \sqrt{(\mathbf{P}^T \mathbf{P})^{-1} \sigma_{\mathbf{m}}^2} \quad (17)$$

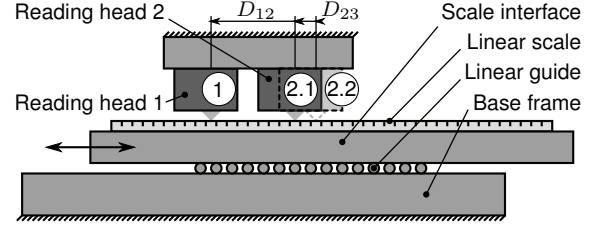


FIGURE 2. Self-calibration concept. The reading heads or the scale are translated over the scale's measurement stroke. Next, Reading head 2 is displaced a distance D_{23} and the scale is scanned again.

in which $\sigma_{\mathbf{m}}$ is the uncertainty of the measurements in $\mathbf{m}$. This equation is only valid if the measurement errors in $\mathbf{m}$ are uncorrelated, which would only be the case for measurement noise, but not for errors like Abbe-errors or thermal errors. For this reason, and in order to validate the calibration method, a Monte Carlo simulation was set up.

In the simulation, a random length deviation profile $\mathbf{e}^*$ was created with typical values of the deviation that are based on the calibration chart of the scales, supplied by the manufacturer. For each experiment in the Monte Carlo simulation, a virtual measurement of the displacement of the two reading heads was conducted, while for every measurement the following errors were added to the data: Abbe error $e_{\text{Abbe}} = \Delta_{\text{Abbe}} \cdot \Delta\theta_{\text{guides}}$, noise from sensors and system dynamics e_{noise} , deviation of D_{12} and D_{23} and thermal drift e_T of the reading heads. This simulation has been repeated N times and the error with respect to the true deviation has been determined for every calculated deviation. The standard deviation of this error is then calculated for each experiment and the uncertainty is determined as the mean value of the standard deviation of all these experiments. The latter is only valid if, for each experiment, the mean value of $e(k)$ is zero, which is true because of (8).

The simulation was carried out for an encoder with a measurement length of 170 mm and for the following parameters and error sources:

- $N = 100$
- $D_{12} = 54.85 \pm 0.002$ mm (can be adjusted based on the read-out of the encoder)
- $D_{23} = 3.3 \pm 0.002$ mm
- $\Delta x = 0.05$ mm

- $\sigma_{\text{noise}} = 0.1 \text{ nm}$ (each measurement $x_{i,j}$ is the result of an averaging over multiple samples, which reduces the noise)
- $\Delta_{\text{Abbe}} = \pm 0.1 \text{ mm}$ (manufacturing tolerances)
- $\Delta\theta_{\text{guides}} = 36 \mu\text{rad}$ (from guide manufacturer's specifications)
- $e_T = \pm 1 \text{ nm}$ (empirical data)
- $\max(e^*) = 0.2 \mu\text{m}$ (based on calibration data supplied by manufacturer)

The true length deviation e^* and the calculated profile e are shown in Figure 3 for one of the experiments. The calibration error for that experiment is shown in Figure 4. From the standard deviation of the errors in these experiments (Figure 5), the uncertainty ($k=2$) of the calibration procedure has been calculated to be 2.6 nm, which would be feasible to reach the required final calibration uncertainty. Note that the high-spatial-frequency error components that are visible in the figure can be low-pass filtered, since it is known that the true deviation does not contain these high-frequency components.

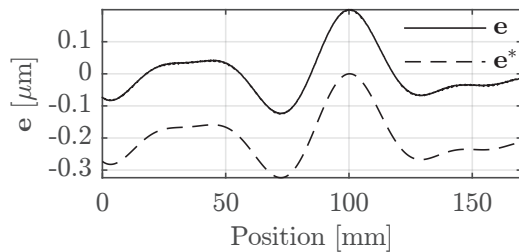


FIGURE 3. Calculated (e) and true (e^*) length deviation of one experiment of the Monte Carlo simulation of the self-calibration procedure for a 170-mm scale. e^* is shifted by $-0.2 \mu\text{m}$ for clarity.

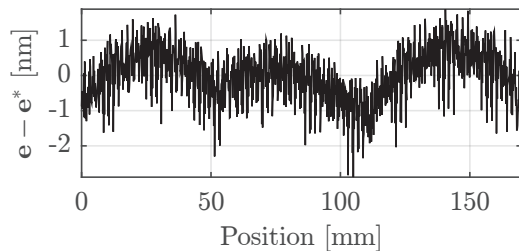


FIGURE 4. Calibration error of one experiment of the Monte Carlo simulation of the self-calibration procedure for a 170-mm scale.

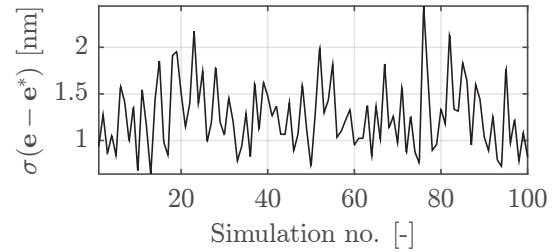


FIGURE 5. Standard deviation of the Monte Carlo simulation experiments of the self-calibration procedure for a 120-mm scale. The calculated uncertainty ($k=2$) is 2.6 nm.

The sensitivity of the calibration procedure to each of the error sources has also been determined by numerical simulation. The width of the probability distribution of each error source has been varied 10 times and the corresponding experiments were run 20 times. The outcome of this simulation gives the uncertainty in function of the range of the error source. The sensitivity values were derived from the first-order-linear-least-squares fitting of the obtained data:

- $\frac{dU_e}{dD_{ij}} = 0.01 \text{ nm}/\mu\text{m}$
- $\frac{dU_e}{d\sigma_{\text{noise}}} = 9.6 \text{ nm}/\text{nm}$
- $\frac{dU_e}{d\Delta_{\text{Abbe}}} = 1.2 \text{ nm}/(100 \mu\text{m})$
- $\frac{dU_e}{de_T} = 1.3 \text{ nm}/\text{nm}$
- $\frac{dU_e}{d\max(e^*)} = 0.73 \text{ nm}/\mu\text{m}$

Because σ_{noise} is an uncorrelated error source, its sensitivity has also been derived from (17) to validate the sensitivity calculated from the Monte Carlo simulation. This gives a value of 9.9 nm/nm, which is fairly corresponding with the value of the simulation.

EXPERIMENTAL VALIDATION

The calibration method has been experimentally verified on a Heidenhain LIP 281 linear encoder with a measurement length of 170 mm that is contained in a setup that was previously used for determining the reproducibility of an Abbe-compliant linear encoder-based measurement system [5]. On this setup, one of the mounts for the reading heads was replaced by a mount on which the reading head can be displaced by set screw, while the motion is guided by two parallel leaf springs. The maximum allowable stress in the leaf springs

limits the maximum displacement of the reading head to 0.3 mm, which puts a limit on the attainable calibration uncertainty.

The calibration procedure has been carried out 9 times on this setup over a length of 146 mm and the resulting calculated deviations e_i are depicted in Figure 6, together with their mean value $\bar{e}$. The difference with respect to this mean value is shown in Figure 7. The periodicity can be ascribed to the amplification of certain spatial frequency errors by the matrix multiplication of (15). From the difference in calculated length deviations, the reproducibility $U_{\text{cal,rep}}$ ($k=2$) of the calibration procedure is estimated at 15 nm. The reproducibility incorporates the following previously simulated errors: deviation of D_{12} and D_{23} , noise e_{noise} , non-repeatable Abbe-errors $e_{\text{Abbe,nr}} = \Delta_{\text{Abbe}} \cdot \Delta\theta_{\text{guides,nr}}$ and thermal drift e_T of the reading heads. Consequently, it can be assumed that only the repeatable Abbe errors are not included in the measurement error and hence are not contributing to the reproducibility. If we assume a maximum Abbe-offset of 100 μm and we calculate the sensitivity to Abbe-offsets for this setup, which equals 2.9 nm/(100 μm), the total uncertainty equals

$$\sqrt{U_{\text{cal,rep}}^2 + U_{\text{Abbe,rep}}^2} = 16 \text{ nm.}$$

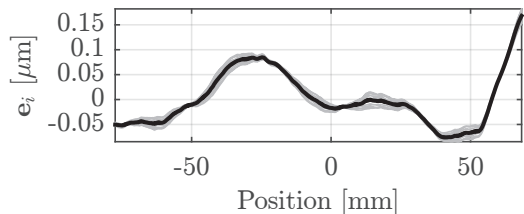


FIGURE 6. Length deviation e_i for 9 self-calibration experiments conducted on a Heidenhain LIP 281 scale over a measurement length of 146 mm.

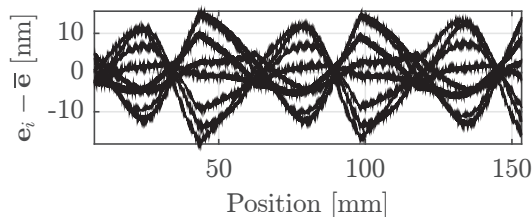


FIGURE 7. Difference of the length deviation e_i with the mean length deviation $\bar{e}$.

The Monte Carlo simulation was used again to

verify whether the estimated calibration uncertainty of the setup corresponds with the simulated uncertainty. Instead of generating random length deviation profiles, the previously estimated profile of Figure 6 was implemented. The other parameters used in the simulation were: $N = 100$, $D_{12} = 54.85 \pm 0.002 \text{ mm}$, $D_{23} = 0.3 \pm 0.002 \text{ mm}$, $\Delta x = 0.05 \text{ mm}$, $\sigma_{\text{noise}} = 0.1 \text{ nm}$, $\Delta_{\text{Abbe}} = \pm 0.1 \text{ mm}$, $\Delta\theta_{\text{guides}} = 36 \mu\text{rad}$ and $e_T = \pm 1 \text{ nm}$. This gave a calculated calibration uncertainty of 18 nm, which is in line with the estimated value of the measurement results. From this we can conclude that the simulation can be used to predict the measurement uncertainty of an improved self-calibration setup in the future and that the simulation results for the 170-mm scale of the previous section are reliable.

The calibration uncertainty for the current setup is mainly caused by a high sensitivity to thermal drift dU_e/de_T of 16.7 nm/nm due to a small displacement D_{23} of 0.3 mm. If the setup would allow an increase of D_{23} to 3.3 mm, the sensitivity would drop to 2.0 nm/nm, which results in an estimated uncertainty of 3.5 nm.

CONCLUSION

This paper presented a method for the self-calibration of the non-linear length deviation of linear encoders by using two reading heads. The method allows calibration of a scale while it is mounted on the machine by adding a second reading head which can be translated over a small distance. Simulations and experiments have shown that the calibration uncertainty decreases with increasing displacement. For a displacement of 3.3 mm, it has been calculated that the calibration uncertainty ($k=2$) for a 170-mm scale can be as low as 2.6 nm. Experimental results have verified that the numerical simulation of the procedure can give an indication of the calibration uncertainty.

REFERENCES

- [1] Kunzmann H, Pfeifer T, Flügge J. Scales vs. Laser Interferometers Performance and Comparison of Two Measuring Systems. CIRP Annals - Manufacturing Technology. 1993;42(2):753 – 767.
- [2] Geckeler RD, Fricke A, Elster C. Calibration of angle encoders using transfer functions. Measurement Science and Technology. 2006;17(10):2811.

- [3] Geckeler RD, Link A, Krause M, Elster C. Capabilities and limitations of the self-calibration of angle encoders. *Measurement Science and Technology*. 2014;25(5):055003.
- [4] Lu XD, Graetz R, Amin-Shahidi D, Smeds K. On-axis self-calibration of angle encoders. *CIRP Annals - Manufacturing Technology*. 2010;59(1):529 – 534.
- [5] Bosmans N, Qian J, Reynaerts D. Reproducibility and dynamic stability of an Abbe-compliant linear encoder-based measurement system for machine tools. In: *Proc. of the ASPE 2014 Annual Meeting*; 2014. p. 51–55.

EXPERIMENTAL VERIFICATION OF ALIGNMENT RELATED ERRORS IN POSITION METROLOGY OF AXES OF ROTATION

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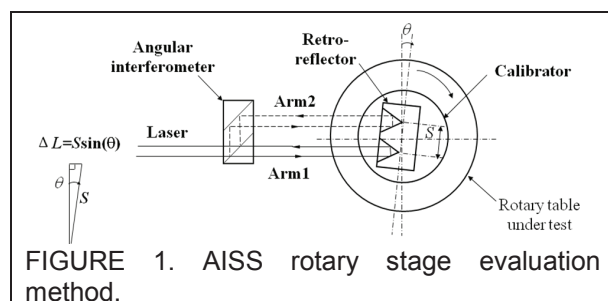
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INTRODUCTION

Calibration of the position accuracy of axis-of-rotation stages is part of a general performance evaluation of computer numerically controlled machining centers and lathes. (Inter)national standards have been written to facilitate these evaluations, e.g. ISO 230-Test code for machine tools. This paper concerns the testing of such axes of motion and the errors incorporated in the data due to misalignment of optical components and devices used for such testing. These errors compromise the uncertainty of the measurement data that could be attained if the alignments were optimal

Specifically, this report addresses issues relating to techniques and instruments which utilize an angular interferometer with stacked (rotating) stages (AISS) [1] as in FIGURE 1. The calibrator may be an automatic rotary stage calibrator (ARSC) or manual rotating stage (MRS). Likewise the rotary stage may be CNC (computer numerically controlled) or a MRS. Expected measurement errors due to misalignments in angular interferometer systems were reported on by Bryan [2] and for rotary stages summarized by Chapman [3]. These types of errors are examined here for AISS metrology methods.



An AG Davis one degree incremental calibration stage (DCS) was first co-evaluated for performance using an AISS method with a secondary manual rotating stage (MRS) and then measured with an ARSC. These measurements provided a baseline evaluation so that the rotary axis of a computer numerically controlled CNC machine tool could be evaluated and then used as a testbed to confirm the magnitudes of error effects related to calibration alignment issues using the AISS method.

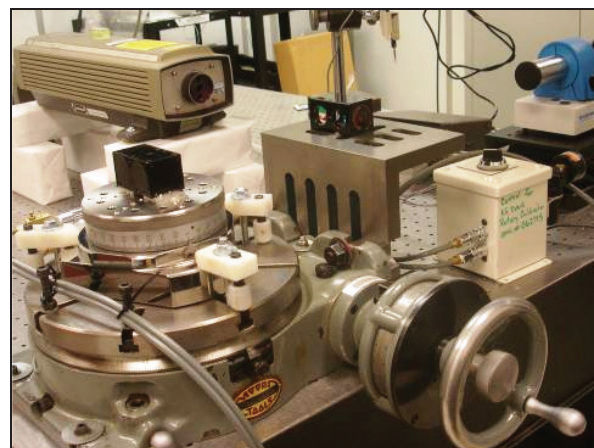


FIGURE 2. Assessment of AG Davis calibration stage using stacked stages and an angular interferometer.

AISS MRS METHOD

Indications from an angle interferometer provided positioning angle displacement information while sequentially rotating the DCS stage at +5 degree increments interleaved with a manual rotation stage rotation at -5 degree increments. (see FIGURE 2)

Subsequently the displacement for each step was calculated and then the positioning summed for each stage step allowing the determination of the 5 degree positional errors of each stage. The sum of the positioning stepping was forced to be $\pm 360^\circ$ over one complete revolution for each rotary stage. The average calculated accuracy of the Davis stage is shown as a dashed line in FIGURE 3.

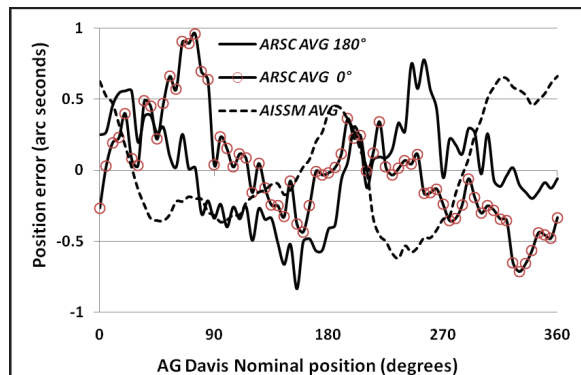


FIGURE 3. Measurement of the errors of DCS by AISSM and using the ARSC.

In a similar setup an ARSC was positioned on top of the DCS and measurements were taken using an automated system. The ARSC system has a built in rotation stage whose position is determined by an calibrated encoder providing the counter-positioning to keep the angular interferometer at a nominal 0 degree position. The stated accuracy of the ARSC was 1 arcsecond which is confirmed by the comparison between the two systems. Additionally the ARSC was mounted in a reverse orientation to see if there were any observable effects. The evaluations from the different orientations are also shown in FIGURE 3. So we see that the accuracy is confirmed regardless of the orientation. Especially considering that the repeatability for the datasets of either method showed a repeatability of about 0.5 sec

BASELINE MEASUREMENT FOR ALIGNMENT ERROR COMPARISON

In order to facilitate automatic measurements in a real environment a rotary axis of a Haas TM-1 CNC machining center that was about 10 years old was utilized.

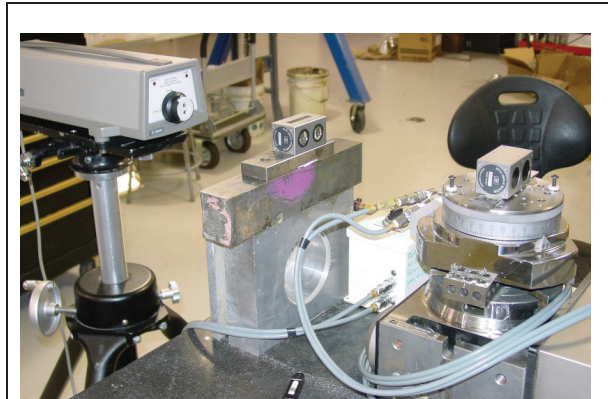


FIGURE 4. AISSM method using AG Davis calibrator on Haas CNC rotary stage with HP (now Keysight) angular interferometer optics.

The baseline set of measurements was taken on the Haas using good metrology procedures paying attention to the alignments. Both the DCS and the ARSC were used to confirm the baseline of the Haas rotary axis (HRA). The setup using the DCS is shown in FIGURE 4. Note that the DCS base is larger than the HRA and 1-2-3 blocks and epoxy were used for mounting. However this degraded the mounting stiffness and the tension on the air connections influenced the stability of the DCS. The average values for the measured accuracy of the HRA using the ARSC are plotted in FIGURE 5. The ± 2 standard deviation regions are plotted for the ARSC and DCS data showing the better stability of the ARSC due to the inferior mounting of the DCS. The ARSC was bolted to the HRA after indicating the center of the mount to within 10 micrometers of the axis of the Haas rotary stage providing better rigidity and repeatability.

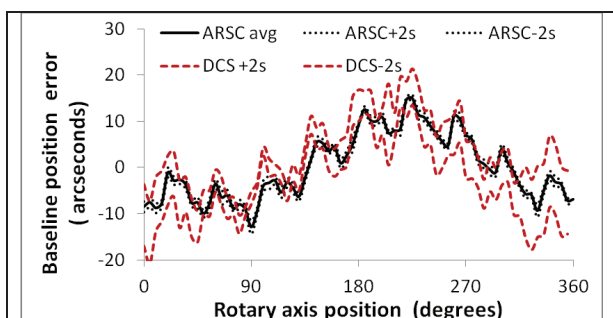


FIGURE 5. Baseline data for Haas rotary axis.

Baseline data is subtracted from data taken with intentional misalignments and compared to the theoretical expectation of the predicted measurement error.

EXPECTED MEASUREMENT ERRORS

If there are angular alignment errors in the measurement loop then there is an error E_m in the measurement of the positioning error described by

$$E_m \approx (\alpha\beta + \omega\eta)\sin(\theta) + (\eta\beta + \alpha\omega)\cos(\theta) + \frac{1}{4}(\alpha^2 + \eta^2)\sin(2\theta)$$

Where :

ω is the roll tilt of the optics on the ARSC with respect to the ARSC frame; α is the pitch tilt and η is the roll tilt of the ARSC axis with respect to the HRA frame; θ is the forward and reverse positioning angles of the ARSC and HRA depending on the operation; β is the misalignment of the laser (tilt of the HRA axis toward the laser beam and ϕ is the initial yaw rotational angle misalignment of the optics with respect to the ARSC frame.

The small angle α ω β and η derivation of the equation can be determined using rotation matrices Q as follows:

$$E_M \approx \begin{bmatrix} 0 \\ 1 \\ 0 \end{bmatrix} \bullet Q_{(\beta)} Q_{(-\theta)} Q_{(\alpha, \eta)} Q_{(\theta)} \begin{bmatrix} 1 - \frac{\phi^2 + \omega^2}{2} \\ \phi \\ \omega \end{bmatrix}$$

with

$$Q_{(\beta)} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 - \frac{\beta^2}{2} & -\beta \\ 0 & \beta & 1 - \frac{\beta^2}{2} \end{bmatrix}$$

$$Q_{(\theta)} = \begin{bmatrix} \cos(\theta) & -\sin(\theta) & 0 \\ \sin(\theta) & \cos(\theta) & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

$$Q_{(\alpha, \eta)} = \begin{bmatrix} 1 - \frac{\eta^2}{2} & 0 & \eta \\ 0 & 1 - \frac{\alpha^2}{2} & -\alpha \\ -\eta & \alpha & 1 - \frac{\eta^2 + \alpha^2}{2} \end{bmatrix}$$

Angles are in units of radians. All third order components of the error were considered to be approximately zero. Also the errors that were dependent on the positioning error itself were ignored as being too small to have an effect in

the second order or in a nonsensitive measurement direction.

MEASUREMENT WITH ALIGNMENT ERROR

First a 0.5 degree tilt/pitch error was introduced in the mounting of the ARSC on the HRA. This was accomplished by placing washers under the ARSC mount on one side. This is seen in FIGURE 6. The washers are facing the laser beam at the initial position. This produces an α error of about 0.5 degrees. This was confirmed by placing a laser pointer in a V-block on the mount with the ARSC removed. The laser spot was marked at some 7 meters away. The DRS was then rotated 180° and also the V-block was rotated 180° and the spot was marked. The vertical separation of the two marks was an indication of the angle. The uncertainty in this angle was estimated to be 0.02 degrees.

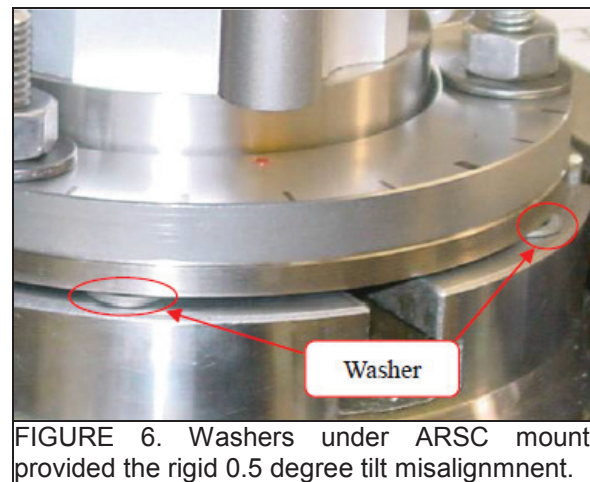


FIGURE 6. Washers under ARSC mount provided the rigid 0.5 degree tilt misalignment.

The laser was also tilted in the vertical plane to an undetermined angle for the measurements. Measurements were taken with the induced tilt and the baseline data was subtracted. The results are shown in FIGURE 7.

The $\sin(2\theta)$ term should have an amplitude of 4 arcsec for $\alpha^2/4$ ($\alpha=0.5^\circ=0.087$ rad). A $\sin(2\theta)$ curve fit to the data yielded an amplitude of 3.7 arcsec for an estimated value of $\alpha=0.49^\circ$ assuming η^2 is much smaller term. This is very close to the originally estimated placement. A curve fit to the sine component yielded an amplitude of $\alpha\beta=22.4$ arcsec. Assuming $\omega\eta$ is much smaller considering the setup, then the laser tilt $\beta=0.7^\circ$ is determined. A $\cos(\theta)$ curve fit yielded an amplitude of 1.4 arcsec so that an

estimated $\eta=0.03^\circ$ (0.5 mrad) if $\omega=0$ or $\omega=0.045^\circ$ (0.8 mrad) if $\eta=0$. A value of $\eta=0.03^\circ$ would correspond to a misalignment of the washers by 4° , which is not unreasonable considering they were positioned by 'eye' in the general direction of the laser.

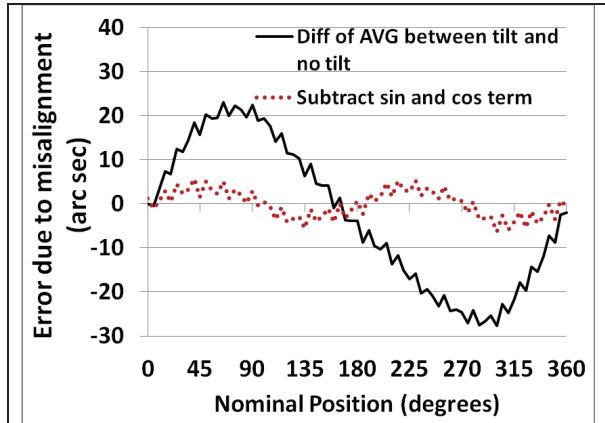


FIGURE 7. Measurement effects due to misalignment of the ARCS and the laser.

The setup was adjusted so that the alignment error was in the roll direction. This also changed the pitch angle of the laser since an optic had to be moved. In this case $\eta=0.5^\circ$. The data is shown in FIGURE 8.

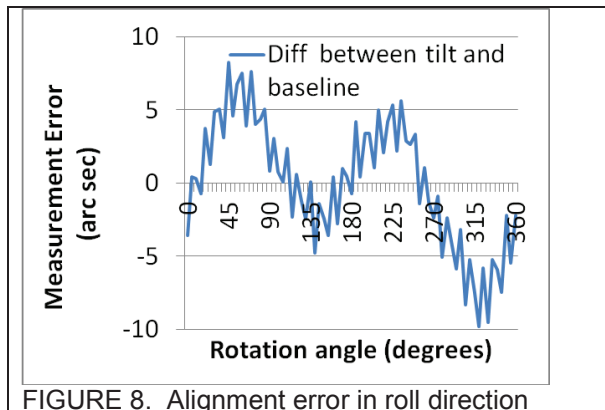


FIGURE 8. Alignment error in roll direction

Similar fits and calculations as before yielded $\eta=0.53^\circ$, $\beta=0.05^\circ$ (a nominal attempt was made to have the laser aligned) and $\omega=0.065^\circ$ (1.1 mrad, reasonable) (alternatively α could have been 0.7 degrees but this result was considered unreasonable)

Another measurement was taken with the tilt(α) angle at an estimated 0.25° . The data is shown

in FIGURE 9. Similar calculations after fits to $\sin(\theta)$, $\cos(\theta)$, and $\sin(2\theta)$ yield estimates of $\alpha=0.21^\circ$, $\beta=0.006^\circ$ (0.1 mrad) and $\omega=0.12^\circ$ (2 mrad).

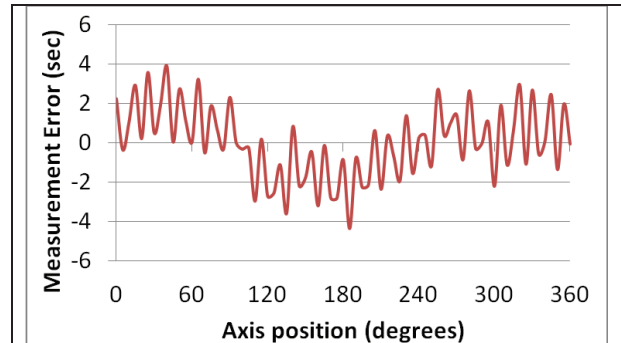


FIGURE 9. Error of measurement with a 0.25° tilt angle

OFF AXIS MEASUREMENTS

Occasionally, due to the construction of a machine it is not possible to place the axis of rotation of the ARSC coincident with the axis of rotation of the rotary stage to be tested. The offset distance may be many tens of centimeters. In this case a secondary linear axis is needed to keep the interferometer optics aligned sufficiently to continue to get a reading. Such a setup is shown in FIGURE 10.

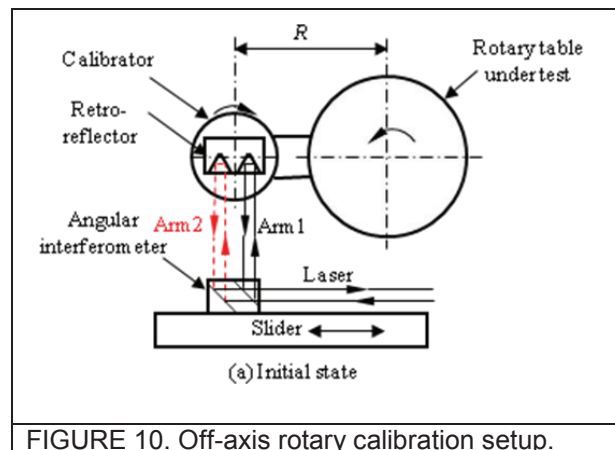


FIGURE 10. Off-axis rotary calibration setup.

This setup introduces the angular errors of the slider into the rotary axis calibration measurements increasing the uncertainty of the measurements. Any tilt in the sensitive direction of the optics shows up in the measurement. See FIGURE 11. If the errors of the slider are measured and subtracted from the results this uncertainty can be reduced but not eliminated due to the non-repeatability errors for each individual measurement.

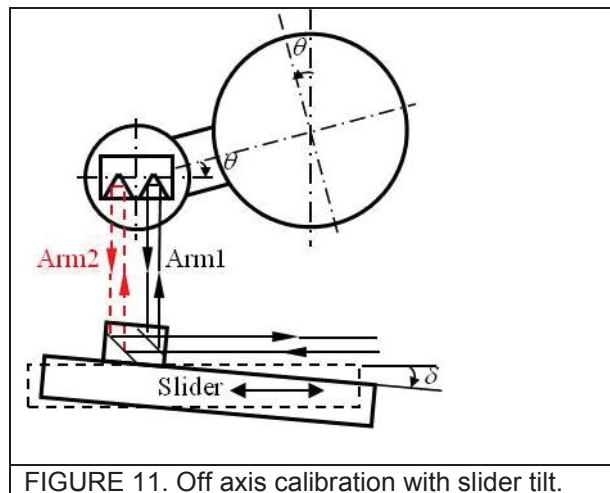


FIGURE 11. Off axis calibration with slider tilt.

The HRA was measured using a linear axis of the Haas machine to provide the slider motion for the optics. A CNC program was written to move the slider and rotary axis at the same time so that there would be no break in the optical signal. Additionally the average measured yaw of the slider was subtracted from the data to see how well it agreed with the baseline data. This data is shown in FIGURE 12.

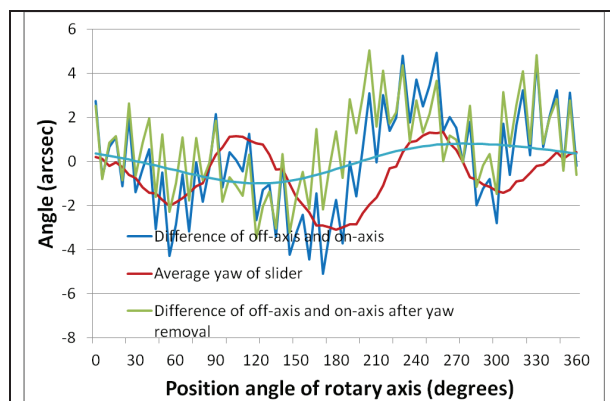


FIGURE 12, Removal of yaw error from the difference between off-axis data and baseline data. The smooth sinusoidal type line is the sum of the $\cos(\theta) \sin(\theta)$ and $\sin(2\theta)$ fits.

An estimate of alignment errors using previous calculation steps showed that they were below a tenth of a degree and would contribute less than 1 arcsec to the data. The result yielded an unaccounted variation in the data on the order of 2 arcsec.

Other possible errors considered were the position difference of the slider and the swing

distance due to the radial offset of the ARSC which could either be in magnitude or offset as shown in FIGURE 13. When coupled with optical misalignment of the interferometer optics they produce a variation $\gamma \propto \cos(\theta)$ [5] for either case. Without apriori knowledge of the error of the optics γ cannot be predicted. But if there is a

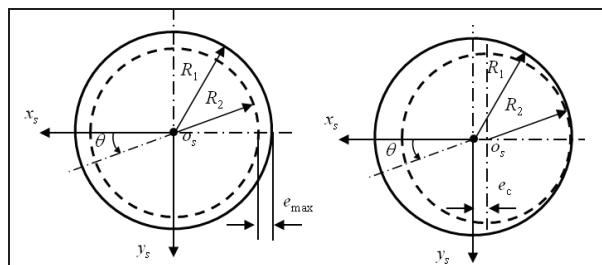


FIGURE 13. Offset and eccentricity alignment errors between the slider and off-axis optics on the rotary axis. The solid line is the contour used for calculating the slider position and the dashed lines are the actual contours.

difference between the cases where there are offsets a cosine variation should be observed. Introducing variations of a 1 mm radial offset e_{max} and then a 1 mm eccentricity e_c revealed little variations of a cosine nature. The data is shown in FIGURE 14. These measurements were not done with any laser tilt so this effect is uncharacterized.

POSITION MEASUREMENT UNCERTAINTY

Because the alignment errors are fixed for a given setup, the terms are estimated to the high side by bounding the alignment errors to a reasonable upper limit. The uncertainty (coverage=2) due to a cosine or sine term is taken to be $\sqrt{2}$. The estimations for each term and the total expected uncertainty (coverage=2) for a typical measurement case is given in **TABLE 1**. Improving the alignments particularly of the laser would, of course, reduce the uncertainty.

SUMMARY

A method for characterizing the position repeatability was analyzed for measurement errors and shown experimentally. In the experimental case given the uncertainty of the measurement should be no greater than 1.5 arc second for an evaluation in the off-axis mode for

the stated position accuracies of the rotary stage and linear slider stage.

ACKNOWLEDGEMENTS

We are grateful to Renishaw for the extended loan of equipment to facilitate portions of this study.

TABLE 1 *Uncertainty estimate (coverage= 2) through bounding the 'fixed' alignments. RSS is the root sum of the squares.*

Component	2*Std-dev or bound	RSS Term
ARSC	1sec 5 μ rad	*
α (tilt between axes)	0.2 mrad 0.01 deg	
η (tilt between axes)	0.2 mrad 0.01 deg	
β (laser tilt)	8.7 mrad 0.5 degree	
ω (optics tilt)	2 mrad 0.1 deg	
$(\eta\beta+\alpha\omega)\cos(\theta)$ $< (\eta\beta+\alpha\omega)\sqrt{2} =$	3 μ rad	*
$(\alpha\beta+\omega\eta)\sin(\theta)$ $< (\alpha\beta+\omega\eta)\sqrt{2} =$	3 μ rad	*
$(\alpha^2+\eta^2)\sin(2\theta)/4$ $< (\alpha^2+\eta^2)\sqrt{2}/4 =$	0.03 μ rad	*
Total on-axis uncertainty by RSS	1.4 sec 7 μrad	*
Off axis contributions		
γ (optics manufacture)	0.1 sec	*
Number of runs	n =4	
Slide yaw repeatability	S _r =1 sec	*
Rotary axis repeatability	R _r =1 sec	
Off-axis deviation $\left[(S_r^2 + R_r^2) / n + \gamma^2 \right]^{1/2}$	0.7 arcsec 3.5 μrad	*
Total uncertainty for off-axis measurement with coverage factor of 2	1.5 sec 7.5 μrad	

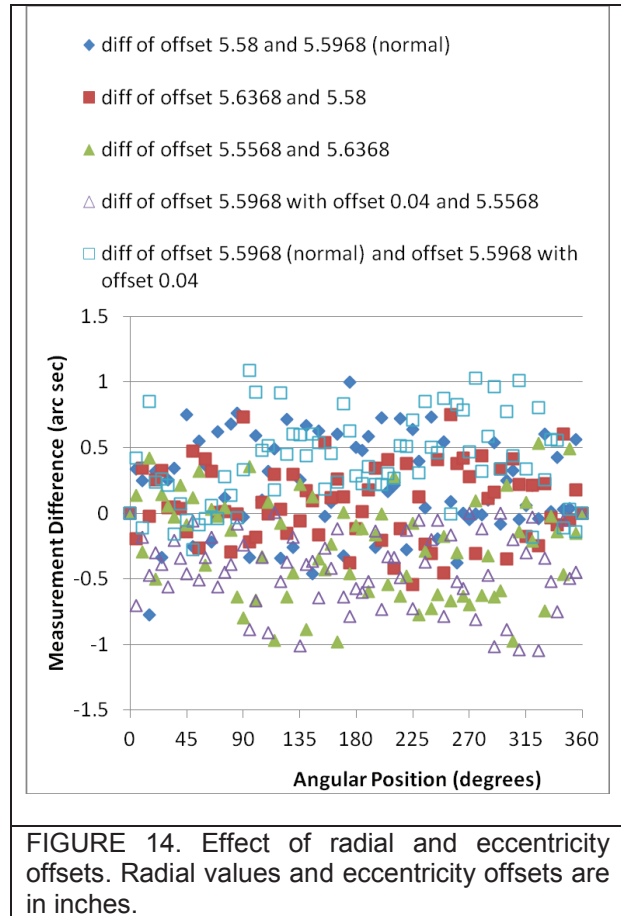


FIGURE 14. Effect of radial and eccentricity offsets. Radial values and eccentricity offsets are in inches.

REFERENCES

- [1] Marzolf JG. Angle Measuring Interferometer. Review of Scientific Instruments. 1964; 35; 1212-1215. DOI:10.1063/1.1719000
- [2] Bryan JB, Carter DL, Thompson SL. Angle Interferometer Cross-Axis Errors. Annals of the CIRP. 1994; 43(1); 453-456. DOI: 10.1016/S0007-8506(07)62251-3
- [3] Chapman MAV, Holloway A, Lee W, May M, McFadden S, Wall D. Interferometric calibration of rotary axes. Available at Renishaw.com
- [4] Zhang G. Improving the accuracy of angle measurement in machine calibration. Annals of the CIRP. 1986; 35(1); 369-70. DOI: 10.1016/S0007-8506(07)61908-8

PASSIVE SEMICONDUCTOR WAFER ALIGNMENT MECHANISM TO SUPPORT INLINE ATOMIC FORCE MICROSCOPE METROLOGY

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ABSTRACT

This paper introduces the design of a mechanical wafer alignment device to enable inline atomic force microscope (AFM) metrology in nanoscale manufacturing by dramatically reducing AFM metrology setup time. The device consists of three pins that exactly constrain the wafer and a nesting force applied by a flexure to keep the wafer in contact with the pins. Kinematic couplings precisely mate the device below a flexure stage containing an array of AFM microchips. The wafer setup time is less than one minute and the lateral positioning accuracy is on the order of 1 μm .

INTRODUCTION

One of the major challenges in nanoscale manufacturing is defect control. Optical inspection is not an option at the nanoscale level due to the diffraction limit of light, and without inspection high scrap rates can occur. One solution to this problem is inline metrology using atomic force microscopes. Single chip MEMS-based AFMs have recently been developed that could be easily and rapidly incorporated into fabrication lines [1]. However, metrology with these AFM chips requires accurate placement of specimens relative to the AFM tip. Further, present set up times for an atomic force microscope are on the order of thirty minutes.

This paper presents a passive wafer alignment mechanism to decrease setup times in AFM metrology. Translational positioning repeatability better than 2 μm and rotational repeatability better than 100 μrad is realized. Sample setup time is reduced to less than one minute. The substantial time savings over traditional alignment methods as well as positioning repeatability well within the range of newly available AFM chips make this mechanism a suitable fixture for inline AFM metrology.

DESIGN AND OPTIMIZATION

A wafer in contact with a flat surface has three degrees of freedom -- two translational and one rotational. Out of plane rotations and translation are thought to be sufficiently limited by the gravitational force on the wafer. Three pin constraints exactly constrain the wafer such that it has zero degrees of freedom. Two pins with intersecting lines of action (LOAs) constrain the translational degrees of freedom of the wafer. A third pin is required to constrain rotation of the wafer and its location is not trivial. Blanding notes that rigid bodies rotate about instantaneous centers located at the intersections of the LOAs of constraints [2]. Thus, in order to constrain a rotational degree of freedom two of the three constraints must be parallel, forming an instantaneous center at infinity and thereby preventing rotation [2]. Conventional wafers possess a flat which provides a convenient location for the constraints with parallel LOAs. A nesting force applied by a flexure mechanism maintains contact between the wafer and the pins.

Determination of the optimal location for the third pin constraint and nesting force is a major objective of this paper. Figure 1 illustrates naming conventions for the geometry of the constraint system. The flexure applies a nesting force F_n at an angle ϕ measured with respect to the x-axis. The pins in contact with the flat of the wafer are referred to as 'left pin' and 'right pin'. The third pin is referred to as 'third pin' and makes an angle θ with respect to the x-axis.

The first step in optimizing the constraint location is analyzing the reaction forces and moments as a function of the unknown angles θ and ϕ . Reaction forces at each pin are determined using matrix inversion to satisfy static equilibrium of forces in the x-y plane and equilibrium of moments about the z-axis [3]. Reaction forces at each pin are shown in Equation 1.

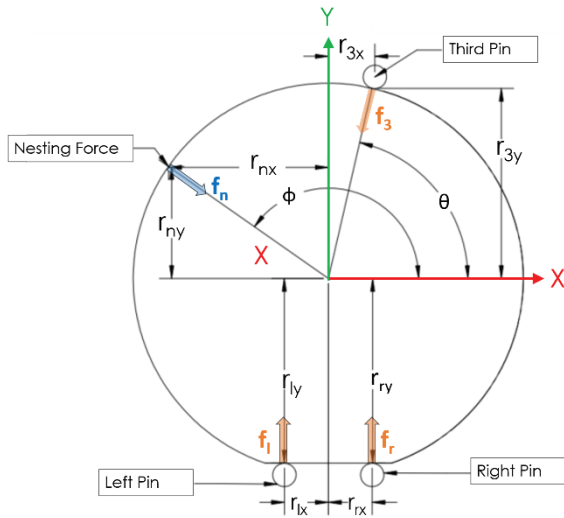


FIGURE 1. Wafer Alignment Naming Conventions

$$\begin{bmatrix} f_3 \\ f_l \\ f_r \end{bmatrix} = f_n \begin{bmatrix} -\cos \phi \sec \theta \\ \frac{1}{2}(\sin \phi - \cos \phi \tan \theta) \\ \frac{1}{2}(\sin \phi - \cos \phi \tan \theta) \end{bmatrix} \quad (1)$$

Allowable combinations of ϕ and θ for a given wafer geometry are those resulting in compressive reaction forces (positive values) as this is a necessity for the wafer to be in static equilibrium. From this constraint we find that ϕ and θ must be on opposite sides of an imaginary vertical line drawn between the left pin and the right pin. We arbitrarily chose the left side of this line for the nesting force and the right side for the third pin location.

Values for ϕ and θ are also limited by a 'Nesting Force Window' [2]. In this paper a numerical method of nesting force window analysis is developed from Blanding's method [2] with the advantage of maximizing the utility of the nesting force. The intersections between pin lines of action constitute wafer instantaneous centers of rotation (ICRs). The moment of the nesting force about each ICR for a wafer in contact with only the pins with lines of action that intersect at that ICR must be in the direction of the pin that is not in contact with the wafer. Figure 2 illustrates the wafer instantaneous centers of rotation as well as the correct orientation of nesting force moments about the ICRs. Equation 2 shows the nesting force moment about ICR1 as a function of θ and ϕ .

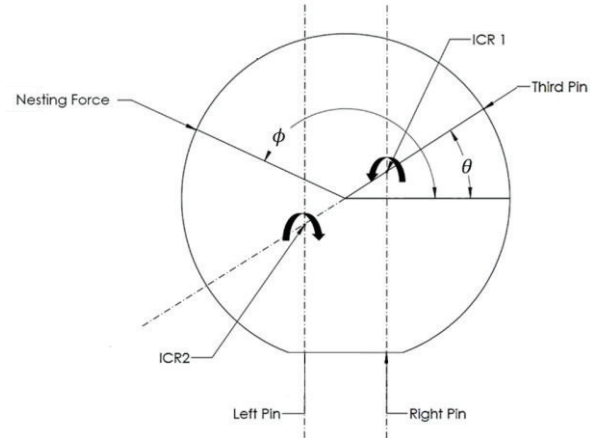


FIGURE 2. Instantaneous Centers of Rotation

$$M_{ICR1} = -\frac{1}{2}F_n(r_{rx} - r_{lx}) * \sec \theta * \sin(\theta - \phi) \quad (2)$$

The nesting force moment is shown to be a function of the magnitude of the nesting force, the distance between the pins located on the wafer flat, and angles ϕ and θ . By symmetry and choice of coordinate system, the two nesting force moments are equal in magnitude and opposite in direction. In order to increase the magnitude of the nesting force moments, the distance between the pins in contact with the wafer flat should be maximized. Maximizing the magnitude of the nesting force would also increase the magnitude of the nesting force moments. The effect of the location of the nesting force and third pin is less clear. A plot of the nesting force as a function of the angles ϕ and θ gives insight into the optimal position of the nesting force and third pin respectively.

The angle between the third pin and the x-axis is bound to the region between -45° and 80° . The lower bound prevents the third pin from interfering with the right pin. The upper bound prevents over-constraint in the y-direction by maintaining a clearance between the vertical coordinate of the top of the wafer and the contact point between the wafer and the third pin. The lower bound for the nesting force angle is 100° to prevent over-constraint and the upper bound for the nesting force angle is 315° in order to prevent interference with the left pin. Limiting the resulting moments to those with the appropriate orientation as shown in Figure 2 allows for the visualization of the nesting force window as a

function of third pin angle θ . The results of this exercise are shown in Figure 3 and it is evident that the restoring moments about the ICRs are maximized as the third pin angle is moved towards its upper bound.

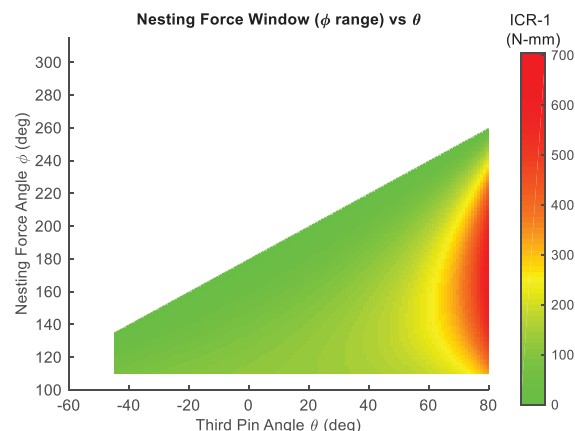


FIGURE 3. Nesting Force Window

PROTOTYPE DESIGN

A prototype alignment mechanism was designed and fabricated to test the repeatability of the optimized design. 5 mm dowel pins were press fit into a block of 6061 aluminum to serve as constraints. The left pin and the right pin were permanently fixed such that they made contact with the wafer at a distance of 4mm from the ends of the 32.5mm flat on the wafer. Holes were drilled for the third pin at angles of -45, 0, 45, 70, and 80 degrees with respect to the x-axis. A prismatic flexure was designed to provide a nesting force of approximately 10N to the wafer at an angle of 135 degrees with respect to the x-axis. The nesting force angle was selected to strike a balance between maximizing the restoring moments about ICRs and generating approximately equal reaction forces at each of the pins. The surface of the alignment mechanism is recessed so that the central axis of the flexure comes into contact with the wafer in order to minimize torsion. Three Vee-Blocks were countersunk into the stage in an equilateral triangle configuration providing a stable kinematic interface for the half-spheres that extended from the mating translation stage. A photograph of the final prototype is shown in Figure 4.

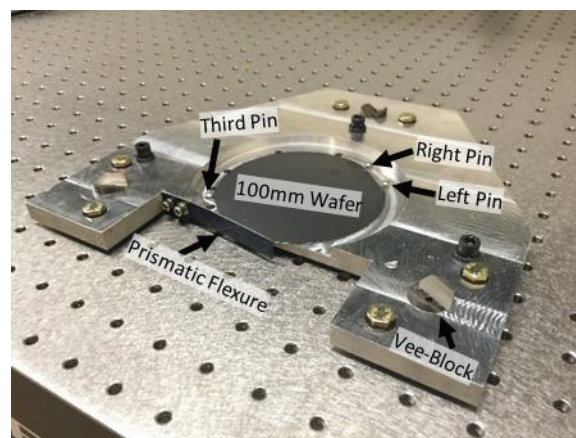


FIGURE 4. Prototype Alignment Mechanism

EXPERIMENTAL SETUP

Experiments were performed to determine wafer placement repeatability as a function of third pin angle θ . Lateral repeatability was determined by measuring the distance from three capacitance probes to a reference block of aluminum bonded to a silicon wafer. Two capacitance probes separated by a known distance and with faces parallel to one face of the block measured translational repeatability in the x-direction and rotational repeatability about the z-axis. Translational repeatability in the y-direction was measured with an additional capacitance probe orthogonal to the two in the x-direction. On the first placement of the wafer, the capacitance probe measured distances were nulled such that each additional measurement was made relative to the first measurement. Between measurements the wafer was completely removed from the stage and then carefully hand-placed back on the stage. For each experiment fifty measurements were recorded with the maximum and minimum values discarded. Repeatability was defined as the standard deviation of the trials. The square root of the sum of the squares of repeatability in the x and y directions was used as a measure of overall lateral repeatability. An image of the experimental setup is shown in Figure 5.

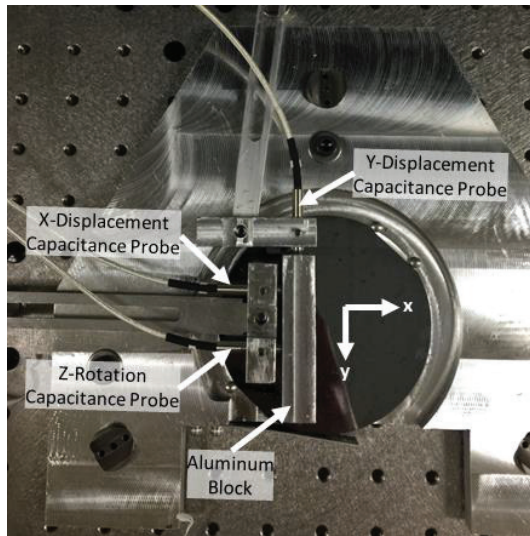


FIGURE 5. Experimental Setup

RESULTS

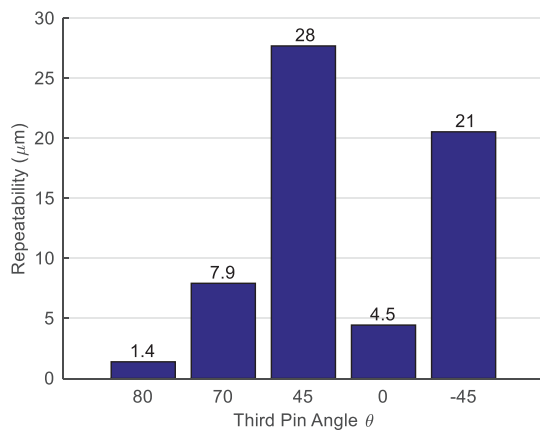


FIGURE 6. Translational Repeatability Results

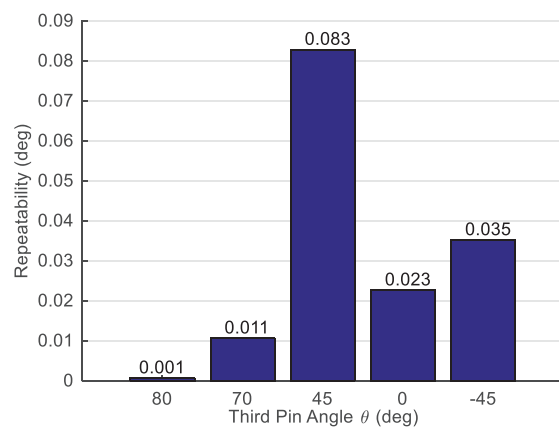


FIGURE 7. Rotational Repeatability Results

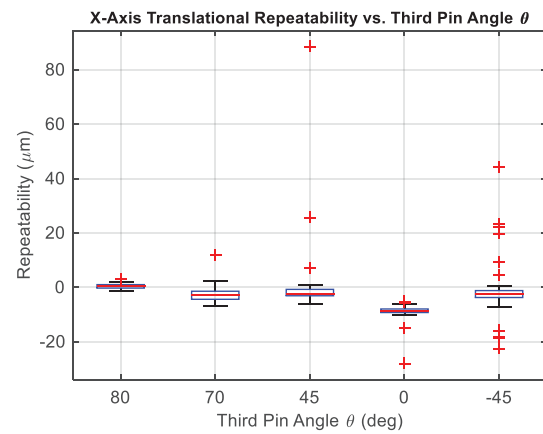


FIGURE 8. X-Axis Repeatability Trials

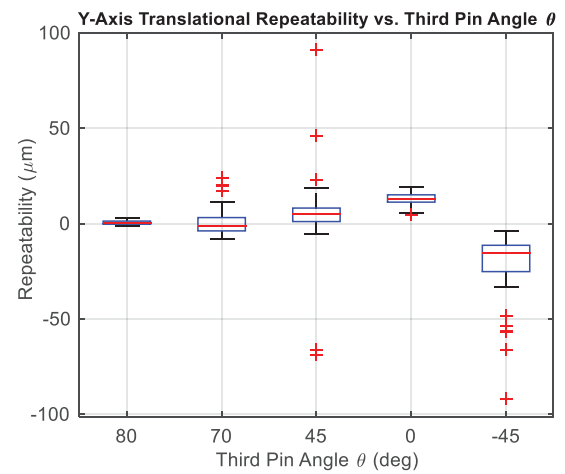


FIGURE 9. Y-Axis Repeatability Trials

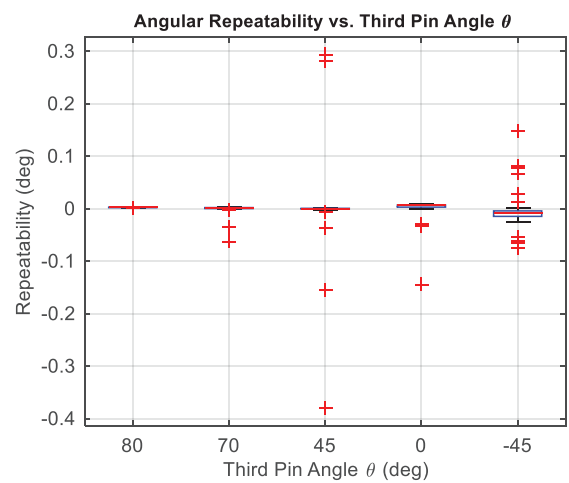


FIGURE 10. Angular Repeatability Trials

Optimal repeatability was achieved with a coupling configuration that maximized the moments about the instantaneous centers of rotation. Positioning the third pin at an angle of 80° resulted in translational repeatability of 1.4 µm. Repeatability of other pin locations was skewed by a number of outliers with positioning errors that are an order-of-magnitude greater than the mean. Figures 8, 9, and 10 give insight into the distribution of the repeatability data. The blue boxes bound results that fall between the 25th percentile (Q1) and 75th percentile (Q3) values of sample data. Median values for each pin location are indicated by a red line within the blue boxes. Statistical outliers are shown as red crosses and are values that are greater than $Q3+1.5*(Q3-Q1)$ or smaller than $Q1-1.5*(Q3-Q1)$. In a number of cases rotation about the x and y axes was observed and an additional experiment will be performed to determine the efficacy of tapering two of the pins to constrain rotational degrees of freedom.

Further trials and analysis will be carried out to determine the effect of varying the nesting force magnitude. Repeatability of the nesting force will also be measured. Preliminary results suggest that 10N is the maximum safe level of force to apply as wafer damage occurred with thicker flexures. Repeatability without the nesting force was challenging to measure as results were often outside of the +/- 125 µm range of the capacitance probes used in the study.

CONCLUSION

The wafer alignment mechanism discussed in this paper is a step forward towards true inline atomic force microscope metrology. Presently it is only possible to scan the surface of wafers with an AFM in a laboratory setting equipped with instrumentation and mechanical stages capable of precisely aligning the wafer relative to the AFM. Alternative passive alignment methods have been proposed which allow sub-micrometer repeatability [4] but these methods require modification of the wafers which is cost prohibitive. Further, advances in AFM technology are lowering the level of precision required for initial alignment to a level which we aim to attain using quasi-kinematic design principles.

The research conducted also serves to advance the science of precision design. Notable experts such as Blanding [2] and Slocum [5] amongst others [6] [7] have laid an excellent foundation for

systematically designing highly accurate and repeatable mechanical systems. The research we have conducted serves as a case study in the application of these methods. The results of our experimentation with various angles for the nesting force and third pin locations will provide insight into how to optimize the design of fixtures with parallel constraints.

References

- [1] Volden T, Hierlemann A, Sedivy J, Barrettino D, Kirstein K-UU, Hafizovic S, et al. Single-chip mechatronic microsystem for surface imaging and force response studies. *Proc Natl Acad Sci* 2004;101:17011–5. doi:10.1073/pnas.0405725101.
- [2] Blanding DL. Exact constraint: machine design using kinematic principles. ASME Press; 1999.
- [3] Beer F, Johnston J, Russell E, Mazurek D, Cornwell P. Vector Mechanics for Engineers: Statics and Dynamics. McGraw-Hill Education; 2012.
- [4] Weber AC. T. Precision Passive Alignment of Wafers 2002.
- [5] Slocum A. Kinematic couplings : A review of design principles and applications Accessed. *Int J Mach Tools Manuf* 2014;50:310–27.
- [6] Hale LC. Principles and Techniques for Designing Precision Machines. Dep Mech Eng 1999;PhD.
- [7] Hammond A. Establishing A Quantitative Foundation for Exactly Constrained Design. All Theses Diss 2003.

The Hybrid Measurement Technique for the PDGEs and PIGEs at the Identical Coordinate System

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INSTRUCTIONS

The performance of machine tools is an important issue in the manufacture of precise and miniature components.^[1,2] Precise tooling is increasingly required for a wide range of applications in the fields of biotechnology, semiconductor technology, and medical optics. The importance of form accuracy is increased in the miniaturization of components, and therefore investigations of performance improvement of machine tools are essential. The performance of machine tools can decrease due to unexpected geometric, thermal, and dynamic errors.^[3-4] Geometric errors concern the relative position and orientation between the tool and the workpiece. Geometric errors are categorized as PDGEs (position dependent geometric errors), and PIGEs (position independent geometric errors).^[5] PDGEs result from the component's imperfection of a linear axis itself, and vary according the command position. Alternatively, PIGEs are caused by assembly imperfections between the linear axes, and are constant regardless of the linear axes' command position. These errors decrease the machine tool positioning, and therefore compensation for geometric errors is necessary in the fabrication of high-quality components. Hence, geometric errors must be analyzed. A range of systems for measuring geometric error have been suggested and developed. Laser interferometer often used to measure the geometric error of machine tools, but only one error can be measured in a single installation, and the roll error cannot be measured. Costly measurement procedures are also required to minimize the effect by Abbe's error on geometric errors. The measurement of PIGEs is a particularly complicated procedure.^[6] The error of a straightness measurement has greater uncertainty because the sensitive direction of a straightness measured does not coincide with the measurement direction. It is also difficult to measure the straightness error because the measurement optics rotate with angular errors.^[7]

A double ball-bar system is widely used to diagnose the effect of geometric errors.^[8]

However, it is difficult to separate individual geometric errors from the measured deviation with this technique. The multi-DOF measurement system is used with capacitive sensors and a measurement target, namely the straightedge. However, the method has problem of high standard uncertainty of measured errors and no identical reference coordinate systems during measurement.^[9] An optimized five-DOF measurement system was developed to minimize the uncertainty of measured geometric errors.^[10] Eleven geometric errors are measured in total, five PDFGEs per axis, and one PIGE in the linear axes (X and Y-axis). Owing to structural limitations, this system cannot measure the linear displacement error for each axis. The multi-DOF error of a linear air-bearing stage is measured using two instruments.^[11] The geometric errors expect a linear displacement error are measured by an autocorrelator, the interferometer's error measurement kit, and capacitive sensors. Most measurement systems cannot measure all errors of a three-axis machine tool. Also, the use of multiple measurement systems can make it difficult to define errors in a single coordinate system. It is therefore difficult to identify all geometric errors because the translational errors are directly affected by the Abbe's offsets between each coordinate system.

Here, hybrid measurement techniques are suggested to investigate all geometric errors of a three-axis machine tool in a single coordinate system. The five-DOF measurement system is used to measure the five PDGEs for each axis, and three PIGEs. Three PDGEs (linear displacement error for each axis) are measured using laser interferometer, and the optimal Abbe's offsets between the two measuring coordinate systems are estimated through a mathematical approach based on a least-squares method. The linear displacement error is calculated by eliminating the Abbe's errors such that all geometric errors of a three-axis machining tool can be defined in a single coordinate system. Using this method, the geometric errors are investigated to

quantitatively evaluate the performance of a three-axis machine tool.

HYBRID MEASUREMENT TECHNOLOGY FOR ALL GEOMETRIC ERROR

The suggested measurement system is used for the five-DOF measurement system^[10] and laser interferometer. The five-DOF measurement system uses five capacitive sensors and an L-type reference mirror. The five PDGEs – except the linear displacement error for each axis – and the three PIGEs were measured. The linear displacement error was measured using interferometry, and as such, all geometric errors were measured.

Five-DOF Measuring System

The measurement procedure for the fifteen geometric errors of a three-axis machine tool is shown in *FIGURE 1*. First, the five PDGEs for each axis (X, Y), and a PIGE between the X and Y-axis were measured. The PDGEs for each axis (X, Z) and two PIGEs were then measured through tests in the XZ-plane. However, the measured data was not defined in a single coordinate system owing to configuration errors (Θ_y , Θ_z) in the L-type reference mirror. These configuration errors affect the straightness, which indicates the increasing or decreasing values which are the measured data from sensors. To define the measured data in a single coordinate system, the configuration errors were calculated by matching with the X-axis straightness error, and a homogeneous transform to match the measured X-axis of each coordinate system using the calculated divergence of two straightness errors, as shown in Equation (1). The measured X-axis coordinate systems $\{M_{XY}\}$, $\{M_{XZ}\}$ for each experimental result were matched to the X-axis of the reference coordinate system $\{R\}$. The Y-axis of each measured coordinate system $\{M_{XY}\}$ is located in the XY-plane of $\{R\}$ for definition of one PIGE (s_{zy}) between the X-axis and the Y-axis, as shown in *FIGURE 2*. Where s_{zy} is the misalignment of the Z-axis about the Y-axis in reference coordinate system $\{R\}$, so that s_{zy} represents the non-orthogonality error between the X and Y-axis in the XY-plane. Similarly, the Z-axis in the measured coordinate system $\{M_{XZ}\}$ represents the two defined PIGEs (s_{xz} , s_{yz}) from the reference coordinate system $\{R\}$. Therefore,

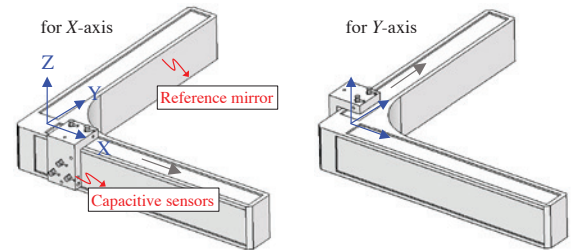
the fifteen geometric errors can be defined from a single reference coordinate.

$$\begin{aligned}\tau_R^X &= \tau_{M_{XY}}^X \mathbf{R}_Z(-\theta_{z,XY}) \mathbf{R}_Y(-\theta_{y,XY}) \\ &= \tau_{M_{XZ}}^X \mathbf{R}_Z(-\theta_{z,XZ}) \mathbf{R}_Y(-\theta_{y,XZ}) \\ \tau_R^Y &= \tau_{M_{XY}}^Y \mathbf{R}_Z(-\theta_{z,XY}) \mathbf{R}_Y(-\theta_{y,XY}) \mathbf{R}_X(-s_{xy}) \\ \tau_R^Z &= \tau_{M_{XZ}}^Z \mathbf{R}_Z(-\theta_{z,XZ}) \mathbf{R}_Y(-\theta_{y,XZ}) \mathbf{R}_X(-s_{xy})\end{aligned}\quad (1)$$

where,

$$\begin{aligned}\mathbf{R}_X(\alpha) &= \begin{bmatrix} \cos \alpha & -\sin \alpha & 0 & 0 \\ \sin \alpha & \cos \alpha & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \\ \mathbf{R}_Y(\beta) &= \begin{bmatrix} \cos \beta & 0 & \sin \beta & 0 \\ 0 & 1 & 0 & 0 \\ -\sin \beta & 0 & \cos \beta & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \\ \mathbf{R}_Z(\gamma) &= \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos \gamma & -\sin \gamma & 0 \\ 0 & \sin \gamma & \cos \gamma & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}\end{aligned}$$

Experiments in XY-plane



Experiments in XZ-plane

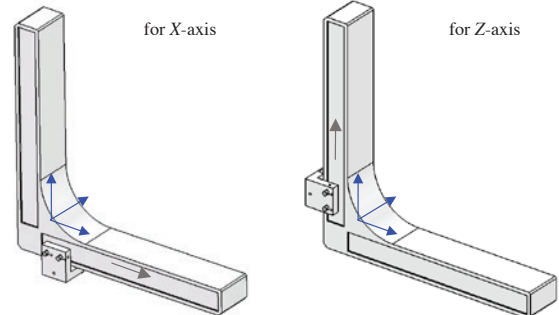


FIGURE 1. Measurement sequence

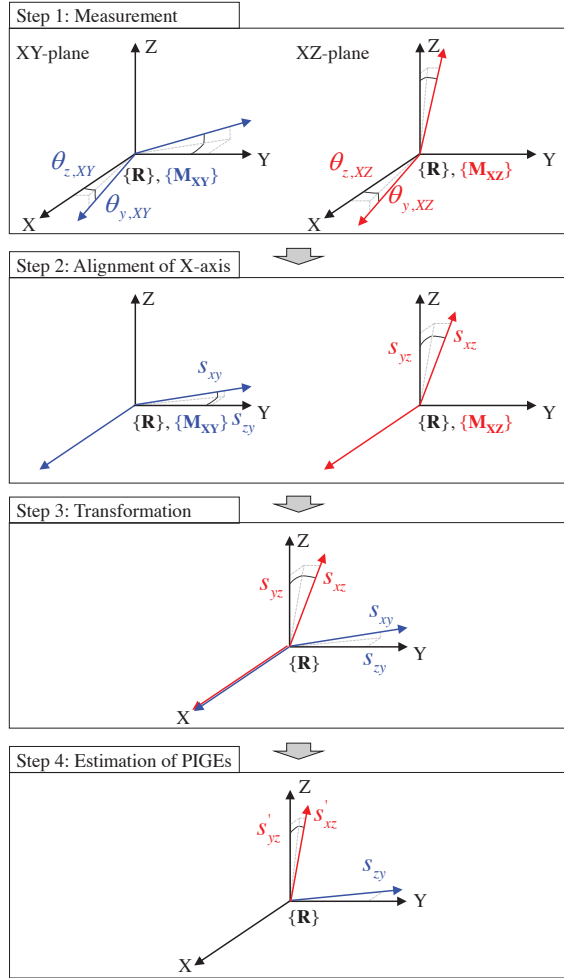


FIGURE 2. The measurement procedure for a five-DOF measuring system

Estimation of Optimal Abbe's offsets

The interferometer was used to measure the linear displacement error, which is not possible with the five-DOF measurement system. However, measured data using independent equipment cannot be easily defined in a single coordinate system; one method for achieving this is proposed here. Three translational errors (one linear displacement error, and two straightness errors) are measured for each axis (X, Y and Z-axis) using the interferometer. However, the measured geometric errors from the interferometer are not defined in the reference coordinate system because of Abbe's offsets (o_x , o_y , o_z). Abbe's offsets inevitably occur during installation when two measurement systems are used to measure the geometric error, and must be eliminated to accurately investigate the geometric error. These offsets affect three translational errors, as shown in FIGURE 3. The 5-DOF measurement system is

defined as the reference coordinate system $\{R\}$, and OM represents the Abbe's offsets between $\{R\}$ and the interferometer's measured coordinate system $\{L\}$. EM_L represents the geometric error measured using laser interferometry with Abbe's offsets. EM_M is the geometric error without Abbe's offsets, measured by the five-DOF measurement system. TM represents the machine instruction for the linear axis driver where $\delta_{ij,k}$, $\varepsilon_{ij,k}$ ($i,j=x,y,z$, $k=m,l$) are the translational (δ) and rotational (ε) errors of the coordinate system $\{j\}$ about the i -axis of a reference coordinate system $\{R\}$, and a measured coordinate system $\{k\}$.

$$\tau_R^E = TMEM_M = OM.TMEM_L OM^{-1} \quad (2)$$

where,

$$TM = \begin{bmatrix} 1 & 0 & 0 & t_x \\ 0 & 1 & 0 & t_y \\ 0 & 0 & 1 & t_z \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

$$EM_M = \begin{bmatrix} 1 & -e_{zi} & e_{yi} & \delta_{xi,m} \\ e_{zi} & 1 & -e_{xi} & \delta_{yi,m} \\ -e_{yi} & e_{xi} & 1 & \delta_{zi,m} \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

$$EM_L = \begin{bmatrix} 1 & -e_{zi} & e_{yi} & \delta_{xi,l} \\ e_{zi} & 1 & -e_{xi} & \delta_{yi,l} \\ -e_{yi} & e_{xi} & 1 & \delta_{zi,l} \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

$$OM = \begin{bmatrix} 1 & 0 & 0 & o_x \\ 0 & 1 & 0 & o_y \\ 0 & 0 & 1 & o_z \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

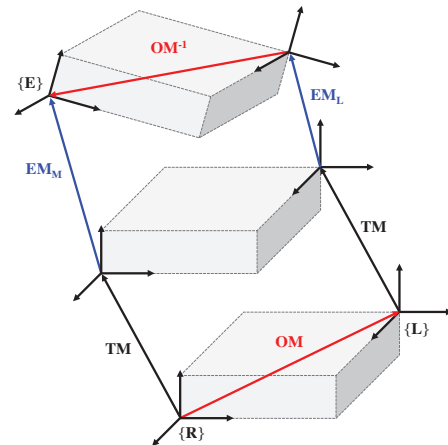


FIGURE 3. Abbe's offsets between two measured coordinate systems

$$\mathbf{EM}_M = (\mathbf{TM})^{-1} \cdot \mathbf{OM} \cdot \mathbf{TM} \cdot \mathbf{EM}_L \cdot \mathbf{OM}^{-1}$$

for *X* – axis

$$\delta_{yx,m} = -\varepsilon_{zx} o_x + \varepsilon_{xx} o_z + \delta_{yx,l}$$

$$\delta_{zx,m} = \varepsilon_{yx} o_x - \varepsilon_{xx} o_y + \delta_{zx,l}$$

for *Y* – axis

$$\delta_{xy,m} = \varepsilon_{zy} o_y - \varepsilon_{yy} o_z + \delta_{xy,m}$$

$$\delta_{zy,m} = \varepsilon_{yy} o_x - \varepsilon_{xy} o_z + \delta_{zy,m}$$

for *Z* – axis

$$\delta_{xz,m} = \varepsilon_{zz} o_y - \varepsilon_{yz} o_z + \delta_{xz,l}$$

$$\delta_{yz,m} = -\varepsilon_{zz} o_x + \varepsilon_{xz} o_z + \delta_{yz,l}$$

$$\begin{bmatrix} \delta_{yx,m} - \delta_{yx,l} \\ \delta_{zx,m} - \delta_{zx,l} \\ \delta_{xy,m} - \delta_{xy,l} \\ \delta_{zy,m} - \delta_{zy,l} \\ \delta_{xz,m} - \delta_{xz,l} \\ \delta_{yz,m} - \delta_{yz,l} \end{bmatrix} = \begin{bmatrix} -\varepsilon_{zx} & 0 & \varepsilon_{xx} \\ \varepsilon_{yx} & -\varepsilon_{xx} & 0 \\ 0 & \varepsilon_{zy} & -\varepsilon_{xy} \\ \varepsilon_{yy} & 0 & -\varepsilon_{xy} \\ 0 & \varepsilon_{zz} & -\varepsilon_{yz} \\ -\varepsilon_{zz} & 0 & \varepsilon_{xz} \end{bmatrix} \begin{bmatrix} o_x \\ o_y \\ o_z \end{bmatrix} \quad (4)$$

$$\delta_{xx,m} = \varepsilon_{zx} o_y - \varepsilon_{yx} o_z + \delta_{xx,l}$$

$$\delta_{yy,m} = -\varepsilon_{zy} o_x + \varepsilon_{xy} o_z + \delta_{yy,l}$$

$$\delta_{zz,m} = \varepsilon_{yz} o_x - \varepsilon_{xz} o_y + \delta_{zz,l}$$

(5)

$\{\mathbf{R}\}$ and the coordinate system $\{\mathbf{E}\}$ are defined as a kinematic chain, given in Equation (2). The relationship between the measured geometric errors from two measurement systems is given in Equation (3). Abbe's offsets were estimated using a least squares method, and the linear displacement error of each axis, as given in Equations (4)–(5). Therefore, all geometric errors of a three-axis machine tool can be defined with a single coordinate system.

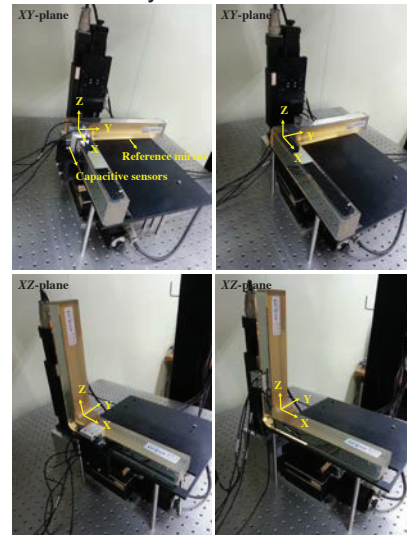
EXPERIMENTS AND EVALUATION

Using the described measurement techniques, all geometric errors of a three-axis machine tool were measured in a single coordinate system. The experiment focused on a three-axis miniaturized machine tool. All geometric errors (three PIGEs and eighteen PDGEs) were measured using a five-DOF system, and a laser interferometer.

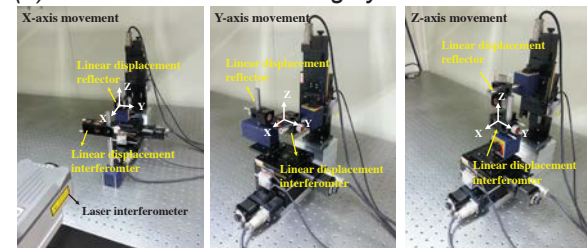
The five-DOS measurement system comprised an L-type reference mirror and five capacitive sensors and was used to measure the five PDGEs for each axis in a single configuration. The five geometric errors of *X* and *Y*-axis were measured under the experiments in the *XY*-plane, and the process repeated for the *XZ*-plane, as shown in FIGURE 4. Each experiment

was repeated five times, and the average of measured results are used. The measured straightness error typically deviated owing to errors in alignments of the L-type reference mirror. Therefore, the method proposed above was used to define the measured geometric errors by two experiments in a single coordinate system.

A least squares method and homogeneous transformation were used to match the two measured coordinate systems. The measured eighteen geometric errors (3 PIGEs and 15 PDGEs) are shown in FIGURE 5. The measured range of each of axes *X*, *Y* and *Z* are 50, 50, and 60mm, respectively, and measurement interval was 10mm. Data were measured in the static state. The linear displacement error of each axis was measured using the laser interferometer, which was stationary after initial installation. The three translation errors for each axis were measured (FIGURE 5), and the optimal Abbe's offsets were estimated, using the method described above, to define the measured errors from laser interferometer in a single coordinate system.



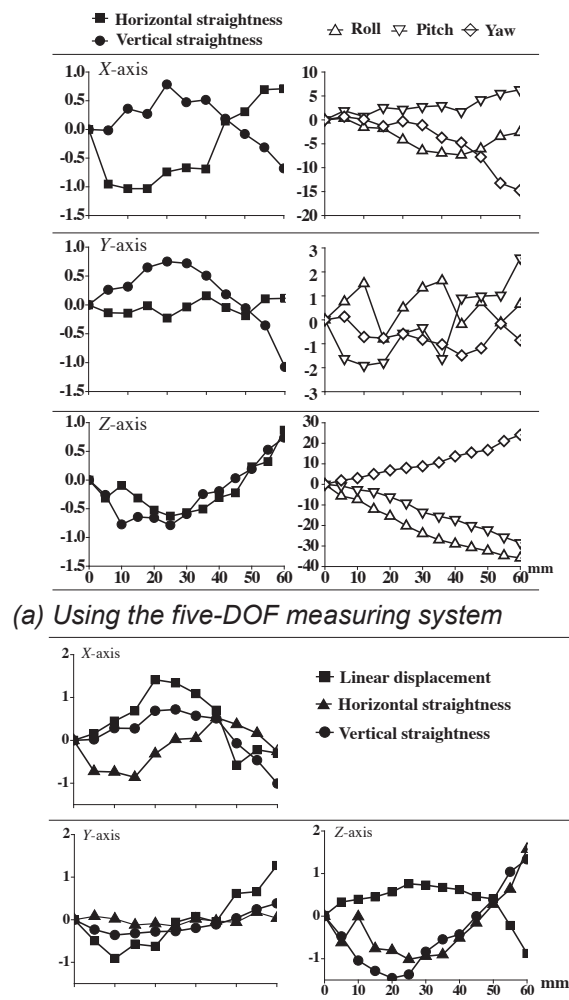
(a) The five-DOF measuring system



(b) The laser interferometer setup for linear displacement error of each axis

FIGURE 4. Experimental configuration for the measurement of geometric errors

The calculated offsets o_x , o_y , o_z were 20.03 mm, -16.12 mm, and 30.80 mm respectively, and the three translation errors without Abbe's offsets are shown in *FIGURE 6*. The straightness errors without Abbe's offset are similar to the experimental results from the five-DOF measurement system. In the Y-axis experimental results, the difference between the linear displacement errors with and without Abbe's offset was 0.4 μm , which was relatively large compared to the maximum linear displacement error of 1.2 μm with Abbe's offset. Using the suggested method, the linear displacement errors without Abbe's offset are calculated and all geometric errors of the three-axis machine tool were defined for a single coordinate system using two measurement techniques. The experimental results are summarized in *TABLE 1*.



(b) Using the laser interferometer
FIGURE 5. Experimental results (units: μm , arcsec)

CONCLUSION

A hybrid measurement technique for the determination of all geometric errors of a three-axis machine tool was suggested. All geometric errors were unified for a single coordinate system by eliminating the effect of Abbe's offsets between two measured coordinate systems. The optimal Abbe's offsets were estimated through an analytical approach, and the linear displacement errors affecting the offsets were calculated to define in a single coordinate system. All geometric errors of an arbitrary multi-linear axis controlled drive were therefore measured for a single coordinate system using the measurement technique developed here. Also these errors, measured by various systems, can be defined the coordinate system using the suggested techniques. The accurately measured errors can be compensated for, improving the performance of a multi-linear axis controlled drive. Additionally, this work can contribute to high precision machining and measurements, thereby enhancing positioning accuracy.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] Bohez E. Five Axis Milling Machine Tool Kinematic Chain Design and Analysis. *International Journal of Machine Tools and Manufacture*. 2002; 42: 505-520.
- [2] Sung-ryung P, Trung-kien H, Seung-han Y. A New Optical Measurement System for Determining the Geometrical Errors of Rotary Axis of a 5-axis Miniaturized Machine Tool. *Journal of Mechanical Science and Technology*. 2010; 24: 337-343.
- [3] Ramesh R, Mannan M, Poo A. Error Compensation in Machine Tools-A Review Part I: Geometric, Cutting-Force Induced and Fixture Dependent Errors. *International Journal of Machine Tools and Manufacture*. 2000; 40: 1235-1256.
- [4] Kiridena V, Ferreira P. Kinematic Modeling of Quasistatic Errors of Three-axis Machining Centers. *International Journal of Machine Tools and Manufacture*. 1994; 34: 85-100.

- [5] Dong-mok L, Zankun Z, Kwang-Il L, Seung-han Y. Identification and Measurement of Geometric Errors for a Five-Axis Machine Tool with a Tilting Head using a Double Ball-Bar. *International Journal of Precision Engineering and Manufacturing*. 2011; 12: 337-343..
- [6] Dong-mok L, Hoon-hee L, Seung-han Y. Analysis of Squareness Measurement using a Laser Interferometer System. *International Journal of Precision Engineering and Manufacturing*. 2013; 14: 1839-1846.
- [7] Gyung-ho K, Tae-ho K, Hu-sang L, Seung-wook K. Compensation of the Straightness Measurement Error in the Laser Interferometer. 2005; 22: 69-76.
- [8] Dong-mok L, Hoon-hee L, Seung-han Y. A Technique for Accuracy Improvement of Squareness Estimation using a Double Ball-bar. *Measurement Science and Technology*. 2014; 25.
- [9] Jae-ha L, Yu L, Seung-han Y. Accuracy Improvement of Miniaturized Machine Tool: Geometric Error Modeling and Compensation. 2006; 46: 1508-1516.
- [10] Kwang-Il L, Jae-chang L, Seung-Han Y. The Optimal Design of a Measurement System to Measure the Geometric Errors of a Linear Axes. *International Journal of Advanced Manufacturing Technology*. 2013; 66: 141-149.
- [11] Wei G, Yoshikazu A, Atsushi S, Satoshi K, Chun-hong P. *Precision Engineering*. 2006; 30: 96-103.

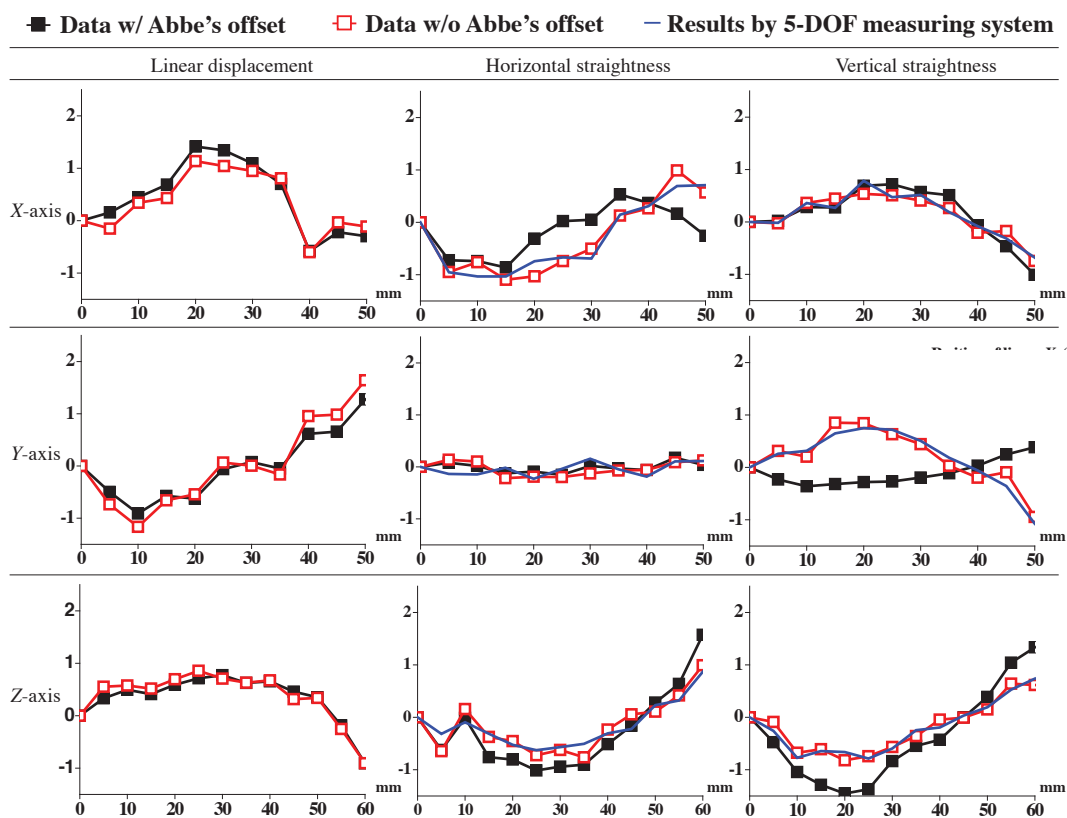


FIGURE 6. Analysis of the experimental results without Abbe's offsets (unit: μm)

TABLE 1. Summary of the experimental results w/o Abbe's offsets (unit: μm , arcsec)

Maximum value of error							
	Linear Displacement	Horizontal Straightness	Vertical Straightness	Roll	Pitch	Yaw	Squareness
X-axis	1.1	-1.0	0.8	-7.3	6.3	-14.7	$s_{zx} : 106.4$
Y-axis	2.1	-0.2	-1.1	1.6	2.6	-1.5	$s_{xz} : 387.8$
Z-axis	-0.9	0.9	-0.8	-35.8	24.2	-28.4	$s_{yz} : -141.0$

CHARACTERIZATION OF DIRECTIONAL FEATURES FOR PRECISION SURFACES USING CURVELET

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Keywords: Curvelet, Directional features, Precision surface measurement, Characterization

INTRODUCTION

Structured surfaces with three dimensional (3D) features such as v-groove [1], pyramid [2, 3] and sinusoid [4] are commonly found in advanced technologies since they have many advantageous specific functions. The dimensional information of the features is incontestably important since they affect the overall optical path. Due to the geometrical complexity of the structured surfaces, it is difficult to evaluate the conformance of this kind of surfaces by most of the optical measurement instruments. This is particularly true when the surfaces are measured and presented in three dimensional data sets.

One of the methods attempts to separate the features per direction and the 3D features are evaluated directionally. Curvelet is the technology firstly developed in image processing [5] which is found to be able to separate directional features including tool marks and scratches [6]. However, the performance is still only limited in a visualization manner.

This paper presents a directional feature evaluation method using curvelet. First, the basic and the detail of the method are provided. Second, experiments are presented for realizing the technical feasibility of the method. The performance of the evaluation method is evaluated quantitatively. Discussion and conclusion with some suggestions for further work are provided at the end. The convective result provides some ideas for the evaluation of the complex surface especially for those with strong directional features.

DIRECTIONAL FEATURE SEPARATION METHOD WITH CURVELET

In this section, the mathematical theory of curvelet is given based on Candes et al. [7]. For the two dimensional space $\mathbf{R}^2$, x is defined as a spatial variable while w is defined as a frequency-domain variable. In the frequency-domain, r is denoted as the radial coordinate while θ is denoted as the angular coordinate for the polar coordinates. When $r \in (1/2, 2)$ and $t \in (-1, 1)$, two windows $W(r)$ and $V(t)$ are defined as the radial window and the angular window. The windows follow the admissibility conditions as shown in Eq. (1) and Eq. (2).

$$\sum_{j=-\infty}^{\infty} W^2(2^j r) = 1, r \in (3/4, 3/2) \quad (1)$$

$$\sum_{\ell=-\infty}^{\infty} V^2(t - \ell) = 1, t \in (-1/2, 1/2) \quad (2)$$

For $j \geq j_0$, the frequency window in the Fourier domain is defined by

$$U_j(r, \theta) = 2^{-3j/4} W(2^{-j} r) V\left(\frac{2^{\lfloor j/2 \rfloor} \theta}{2\pi}\right) \quad (3)$$

where $\lfloor j/2 \rfloor$ is the integer part of $j/2$. Hence, U_j is supported in the wedge which is defined by the support of W and V . By introducing waveform $\varphi_j(x)$ in which Fourier transform $\hat{\varphi}_j(x) = U_j(\omega)$ and the equispaced sequence or rotation angles $\theta_\ell = 2\pi \cdot 2^{-\lfloor j/2 \rfloor} \cdot \ell$ and translation parameters $k = (k_1, k_2)$, the curvelet as function of $x = (x_1, x_2)$ at scale 2^{-j} , orientation θ_ℓ and position $x_k^{(j, \ell)} = R_{\theta_\ell}^{-1}(k_1 \cdot 2^{-j}, k_2 \cdot 2^{-j/2})$ is defined by

$$\varphi_{j,\ell,k}(x) = \varphi_j \left(R_{\theta_\ell} \left(x - x_k^{(j,\ell)} \right) \right) \quad (4)$$

where R_θ is the rotation by θ radians and R_θ^{-1} is its inverse,

$$R_\theta = \begin{pmatrix} \cos \theta & \sin \theta \\ -\sin \theta & \cos \theta \end{pmatrix} \quad (5)$$

Hence, the curvelet coefficient is the inner product of an element f and a curvelet $\varphi_{j,\ell,k}$,

$$c(j,\ell,k) = \langle f, \varphi_{j,\ell,k} \rangle = \int_{R^2} f(x) \overline{\varphi_{j,\ell,k}(x)} dx \quad (6)$$

By Plancherel's theorem, the curvelet coefficient can be expressed as

$$c(j,\ell,k) = \frac{1}{4\pi^2} \int \hat{f}(\omega) \overline{\hat{\varphi}_{j,\ell,k}(\omega)} d\omega \quad (7)$$

The digital form of the curvelet coefficient is

$$c^D(j,\ell,k) = \sum_{0 \leq t_1, t_2 < n} f[t_1, t_2] \overline{\varphi_{j,\ell,k}^D[t_1, t_2]} \quad (8)$$

With the help of curvelet transform, a 2D image can be transformed into a set of coefficients which represent the directional features at different scales. FIG. 1 shows the spatial features corresponding to coefficient at different scales and features. It shows that the curvelet is very suitable for representing directional features.

The block diagram of the proposed directional feature separation method based on curvelet is shown in FIG. 2. Firstly, the original measured data is curvelet transformed using the curvelet toolbox [8], which provides C++ and Matlab version. In this paper, Matlab version is used. After that, the transformation results are separated in several sets of coefficients which represent the directional information of the original surface. According to the desired directions of the surface features, the coefficients can be classified into different groups. Hence, inverse curvelet transformation is performed for different groups of coefficients. Finally, the surfaces with specified directional features are generated. The number of directions can be large. In this paper, only the separation of two directional features is demonstrated.

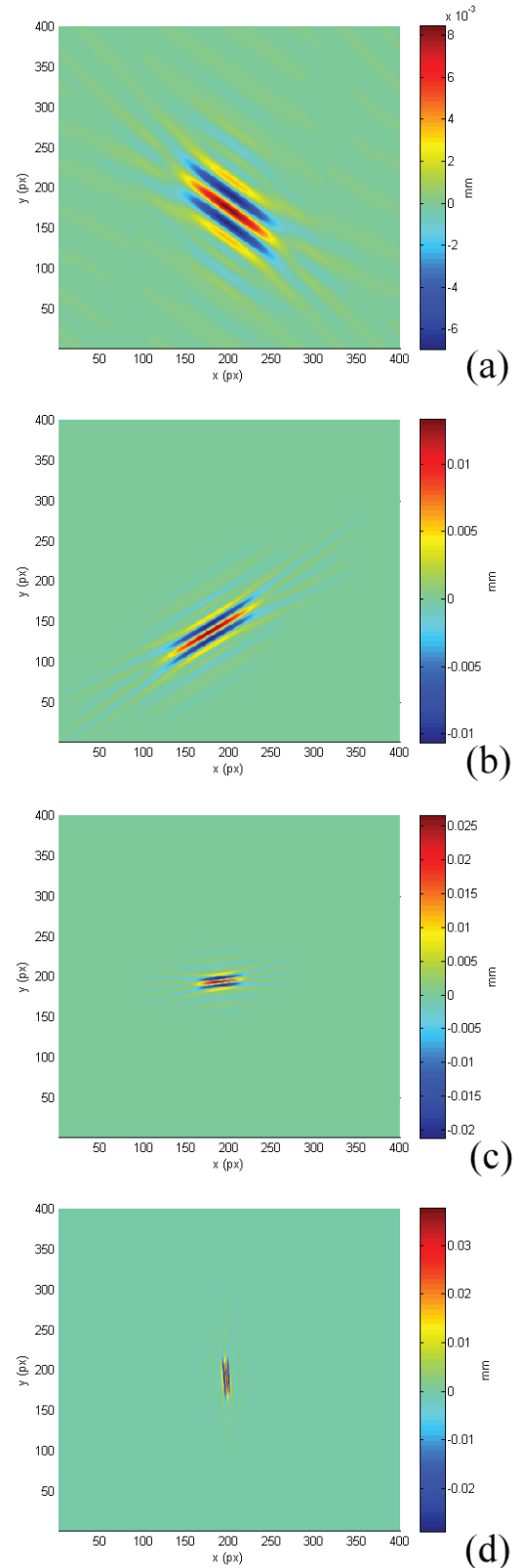


FIGURE 1. Spatial features corresponding to single unit of coefficient at different scales and directions

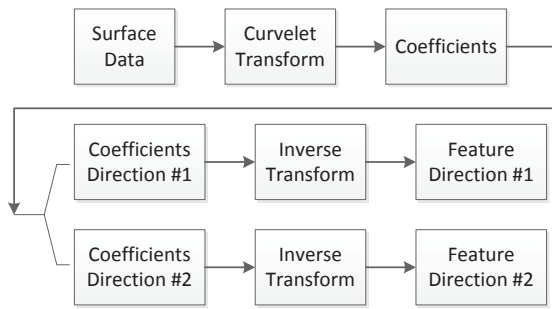


FIGURE 2. Diagram of directional feature separation method

EXPERIMENT VERIFICATION

Sinusoidal Surface

In this section, a Matlab simulation is presented. A simulated sinusoidal surface is shown in FIG. 3 and the surface is designed with Eq. (9).

$$z = \frac{\sin(10x) + 2\cos(5y)}{1000} \quad (9)$$

The equation shows clearly that the simulated surface has two directional features. Both of them are sinusoidal with amplitudes of $1\mu\text{m}$ and $2\mu\text{m}$, as well as frequencies of $\frac{10}{2\pi}$ and $\frac{5}{2\pi}$, respectively.

With the original surface data generated, the next step is to perform curvelet transform. After curvelet transform, the coefficients are calculated and the result is shown in FIG. 4. The coefficients are divided into many sections. In this study, the vertical ones are grouped as A while the horizontal ones are grouped as B.

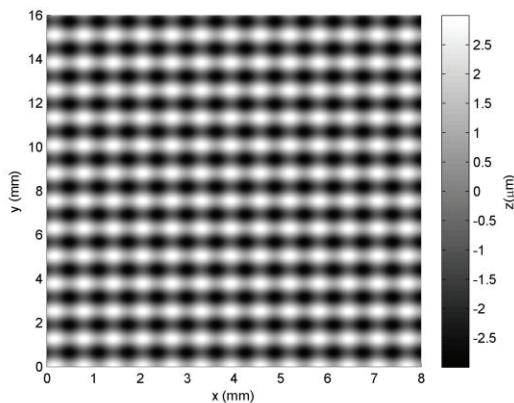


FIGURE 3. Simulated sinusoidal surface

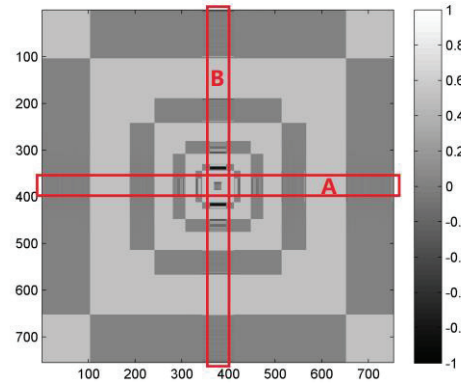


FIGURE 4. Curvelet coefficients and grouping A and B

After performed inverse curvelet transformation with coefficient group A and group B, the reconstructed surfaces are shown in FIG. 5. The result shows that the vertical and horizontal features are well separated.

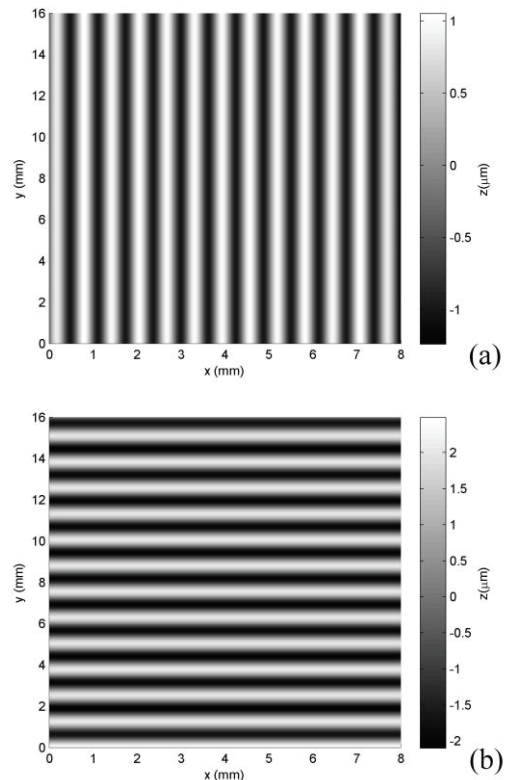


FIGURE 5. Reconstructed surface with directional coefficients (a). Coefficient group A and (b). Coefficient group B

In order to evaluate the performance of this method, the 3D datasets are sectioned into a 2D view. The sectional views of the reconstructed

surfaces and reconstructed error of the profiles are shown in FIG. 6 and FIG. 7, respectively. The results show that the amplitude and the frequency in both directions can be well reconstructed.

However, there are significant distortions in the edge areas while the performance in the middle area is much better, which indicates that the middle part of the result should be used for the characterization. The evaluation error of the profiles in the center part is about $0.1\mu\text{m}$, which indicates the excellent performance of the method.

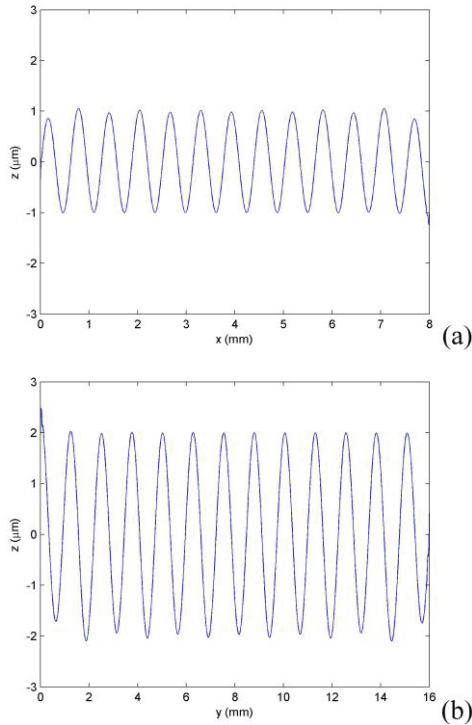


FIGURE 6. Section views of the reconstructed surfaces. (a) x-z view. (b) y-z view

V-groove Surface

Another experiment was performed on a v-groove surface to verify the proposed method. The design parameters of the v-groove surface is summarized in TABLE 1 and it is shown in FIG. 8.

TABLE 1. Design parameters of the v-groove surface

Design parameters	values	units
Cylinder diameter	1000	μm
v-groove width	20	μm
v-groove height	10	μm
v-groove distance	50	μm

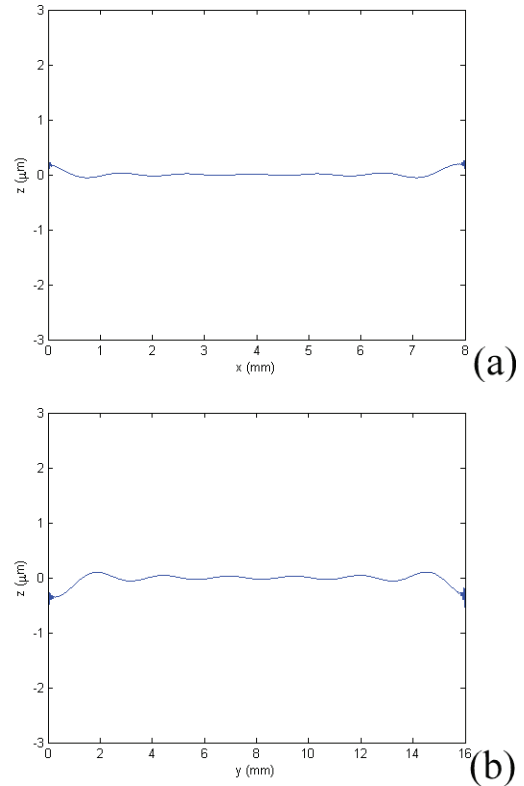


FIGURE 7. Reconstructed error of the two surfaces

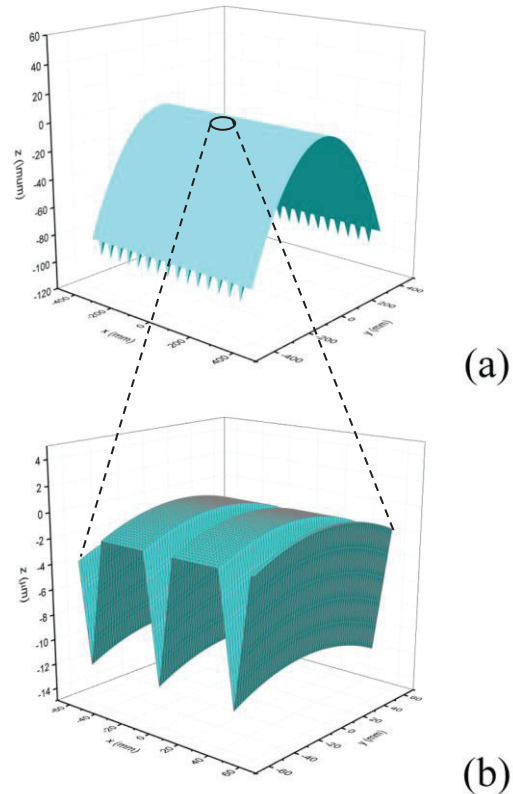


FIGURE 8. Simulated v-groove surface

The section view of the design surface is shown in FIG. 9. One is the main substrates (FIG. 9(a)) and another is the micro groove feature (FIG. 9(b)).

After performed the curvelet transform, the coarsest level and left-to-right coefficients were grouped together and the inverse transform was performed. FIG. 10 shows the result after inverse transform. It is interesting to note that the up-to-down coefficients were not used in this case since there was no up-to-down features in this surface. The coarsest coefficient represents the cylindrical main substrate and it has no directional information.

In order to obtain a better evaluation result, the section views of the reconstructed surface were used to compare with the designed ones. The reconstructed profiles are shown in FIG. 11 and the delta profiles as compared with the designed profiles are shown in FIG. 12. The results show that the evaluation error for the micro-groove is about $2\mu\text{m}$ while the evaluation error for the substrate is about $6\mu\text{m}$. Comparing with the evaluation errors of that for the sinusoidal surface, the results are significant larger, which may be caused by the high frequency components in the v-groove surface. The detail of the influence will be studied in the future work.

CONCLUSION

A directional feature separation method based on curvelet is presented to separate directional features of precision surfaces, which provides a useful characterization solution for the structured surface. Two simulated surfaces with directional features were used to demonstrate the capability of the method. The quantitative analysis of the evaluation performance is also given. The simulation results show that the method is suitable for feature separation. Future work will be focused on the implementation of the method in real surface experiment.

ACKNOWLEDGEMENT

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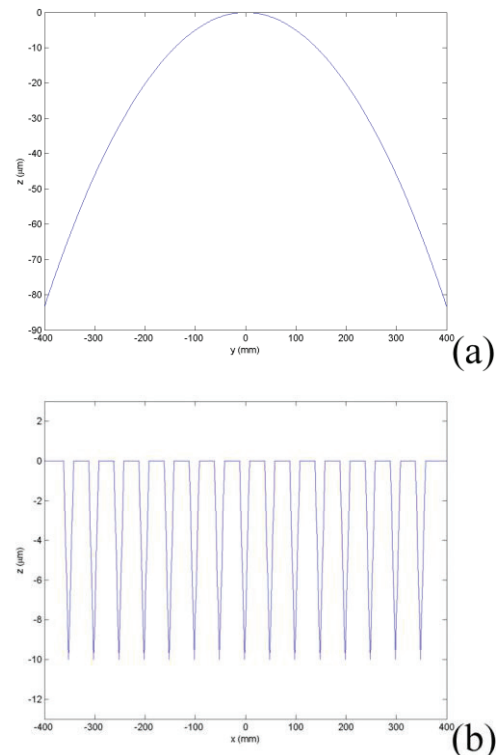


FIGURE 9. Section views of designed v-groove surfaces. (a) y-z view. (b) x-z view

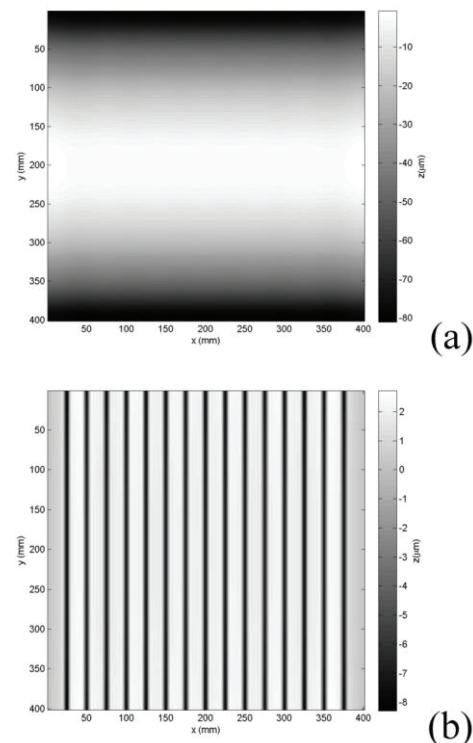


FIGURE 10. Reconstructed surface with different coefficients (a). Coarsest level and (b). Left-to-right coefficients

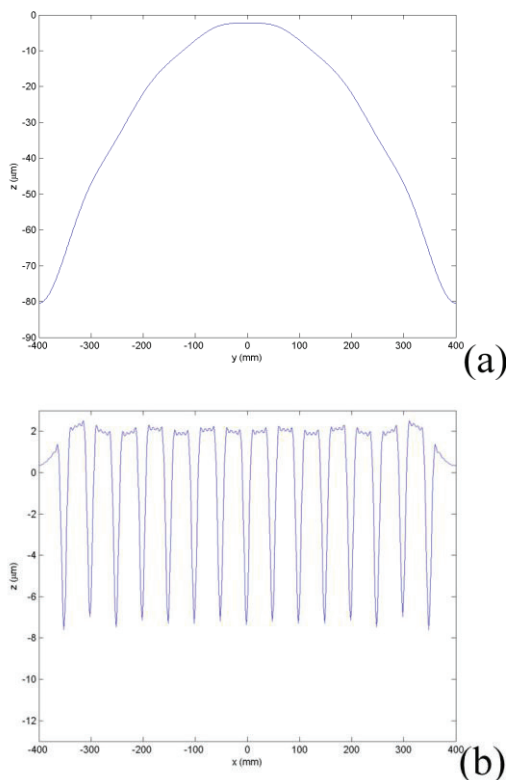


FIGURE 11. Section views of the reconstructed surfaces. (a) y-z view. (b) x-z view

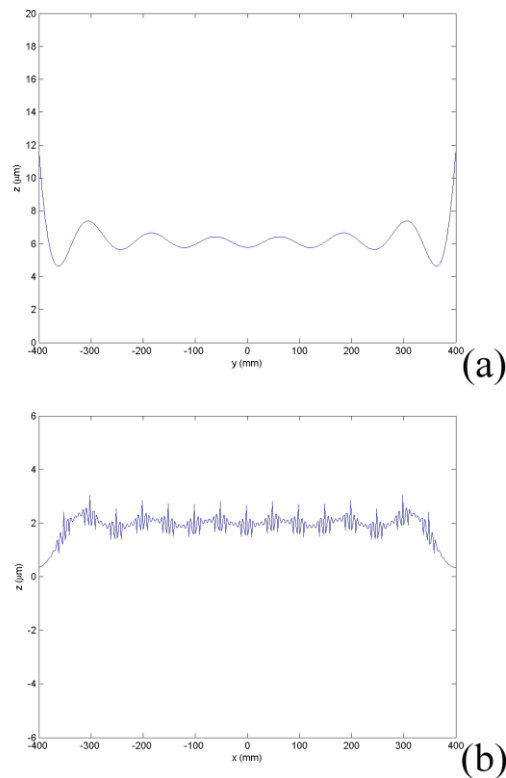


FIGURE 12. Section views of the surface evaluation errors

REFERENCES

1. Bozhevolnyi, S.I., et al., *Channel plasmon-polariton guiding by subwavelength metal grooves*. Physical review letters, 2005. **95**(4): p. 046802.
2. Sodtke, C. and P. Stephan, *Spray cooling on micro structured surfaces*. International Journal of Heat and Mass Transfer, 2007. **50**(19): p. 4089-4097.
3. Dai, H., et al., *Effective light trapping enhancement by plasmonic Ag nanoparticles on silicon pyramid surface*. Optics Express, 2012. **20**(S4): p. A502-A509.
4. Gao, W., et al., *Precision nano-fabrication and evaluation of a large area sinusoidal grid surface for a surface encoder*. Precision Engineering, 2003. **27**(3): p. 289-298.
5. Donoho, D.L. and M.R. Duncan. *Digital curvelet transform: strategy, implementation, and experiments*. in *AeroSense 2000*. 2000. International Society for Optics and Photonics.
6. Li, L., et al., *Feature extraction of non-stochastic surfaces using curvelets*. Precision Engineering, 2015. **39**: p. 212-219.
7. Candes, E., et al., *Fast discrete curvelet transforms*. Multiscale Modeling & Simulation, 2006. **5**(3): p. 861-899.
8. *CurveLab*. 2015 [cited 2015 July 7]; Available from: <http://www.curvelet.org/>.

ERRORS IN SURFACE TEXTURE MEASUREMENTS BY OPTICAL METHOD CAUSED BY NON-MEASURED POINTS PRESENCE

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INTRODUCTION

Surface topography is three-dimensional in nature. Areal (3D) surface parameters are more reliable than 2-D profile parameters. So 3D measurement became popular. The assessment of surfaces using profiles was employed since the early 1930's. The systems basically involved use of the mechanical stylus. There are a lot of factors affecting uncertainty in surface geometry measurement using stylus technique. They are caused by environment, measuring equipment, measured object, software as well as tip shape and size. Errors of stylus method in surface texture measurement are described in book [1]. However, time required to collect real data for areal measurements can appear excessively long compared to the working time of the optical light techniques. This could now become the most important disadvantage of the stylus methods.

Optical methods are fast but are more sensitive to extraneous effects [2]. They are non contacting with more options for improvement than stylus method. From among optical methods, confocal method and white light interferometry are the most popular. Confocal optics are widely applied in engineering and biology because of very high clarity of obtained image but when range is increased the resolution tends to decrease; this factor does not occur when using interferometric methods [2]. Therefore from among optical methods, interferometric techniques are probably the most useful. Coherence scanning interferometry extends interferometric techniques to surfaces that are complex in terms of roughness, steps and discontinuities. Additional benefits include the equivalent of an autofocus at every point in the field of view [3].

The environment can be an important contributor to measurement error in coherence scanning interferometer use. Sharp edges can cause

outliers in the topographic images called spikes [4]. Some peculiarities of the surface caused short time breaks in signal gathered by a photodetector, so there may be some non-measured points. A part of this image can be corrected after using interpolation with an algorithm of determining areas with ambiguous signal. Non-measured points exist also after application of confocal optics.

The fundamental aim of this research was to analyse errors of surface topography measurement caused by non-measured points presence. The other aim was to find surface topography parameters of the smallest and the largest sensitivities on the analysed kind of errors.

MATERIALS AND METHODS

This study is divided into 2 parts. In the first part, circle with non-measured points was artificially created on peak parts of different measured surfaces, with identical measured area and sampling intervals in perpendicular directions. Surfaces were measured by Talyscan 150 stylus instrument with tip of 2 μm radius and by Talysurf CCL Lite interferometer. Measuring area was 3.3 mm x 3.3 mm, sampling interval was 5 μm . The area of this circle was the same for various samples. 10 samples with simulated errors were studied. Non-measured points (of 0.73% NMP ratio) were replaced by a smooth shape calculated from the neighbors using software TalyMap.

In the second part of this study the results of measurement by Talysurf CCI Lite interferometer with 0.01 nm vertical resolution were analysed. Objective 5x was used for all the measurements. Measuring area was 3.3 mm x 3.3 mm, it contained 1024 x 1024 measuring points. The measurements were done with different intensity of light. The results of measurement with smallest number of non-

measured points were the reference data. 15 surface were measured. They were one- and two-process textures isotropic or anisotropic.

THE STUDY OF SURFACES WITH ARTIFICIALLY CREATED NON-MEASURED POINTS

Figure 1 shows examples of the analysed textures. Arrows indicated place of non-measured points presence before filling-in. One can see that corrected part of image is clearly visible in surface after milling shown in Figure 1a, because circle with non-measured points was located in peak part of surface. One can also see the corrected circle in cylinder liner (Figure 1b) and polished surfaces (Figure 1d). Anisotropic character of these textures is the most probably reason of visibility of error (since corrected part is of isotropic character). However it is difficult to find circle with filled-in non-measured points on lapped surface with isolated oil pockets created by abrasive jet machining (Figure 1c). Presence of dimples or/and mixed character of peak surface part after lapping may be the reasons of small visibility of error.

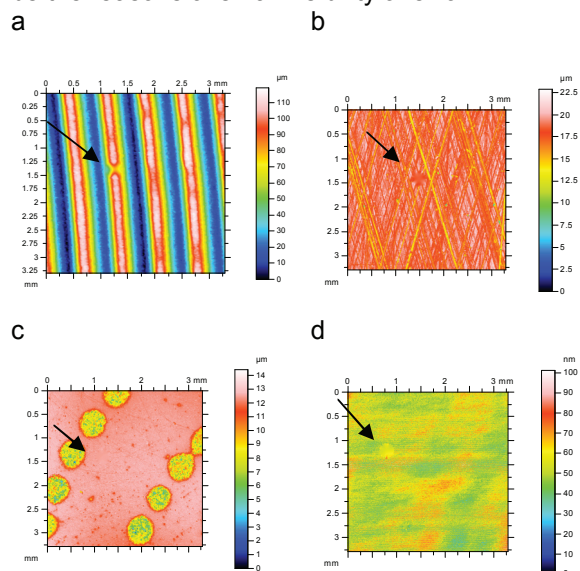


FIGURE 1. Contour maps of surfaces with filled in non-measured points after milling (a), honing (b), texturing by abrasive jet machining after initial lapping (c) and polishing (d).

Relative changes in the surface topography parameters from ISO 25178 standard due to disturbances presences were studied.

Although error caused change of milled surface view (see Figure 1a), relative absolute changes of parameters were not higher than 2%. Different

tendency of various height parameters changes were found; statistical parameters Sa and Sq decreased, as well as Spk and Sv, however the parameters Sk and Sp increased. These changes were not higher than 0.7%. Due to decrease of the Spk parameter as well as of the Sr1 ratio, the Sa1 parameter decreased; its changes were comparatively large (about 1.9%). Both skewness Ssk and kurtosis Sku increased. As the consequence of statistical height parameters decreasing, the hybrid parameters Sdq and Sdr decreased. Material volume Vm increased; its changes were the biggest from all the analysed parameters (about 2%). From among feature parameters, Spd and Sda increased. Spatial parameters were stable. Generally it was found that parameters related to peaks (Sa1 and Vm) changed mostly.

It is interesting that closed dale volume Sdv decreased mostly (about 4%), from among parameters of cylinder liner surface (Figure 1b). Maximum relative change of other parameters was 1.7% (decrease of the Sa1 parameter). From among amplitude parameters only the Sq decreased but Spk increased. Similarly to milled surfaces the Ssk and Sku parameters increased and Sdq as well as Sdr decreased. Functional parameters from V family changed; Vm increased but Vmc and Vc decreased. From among feature parameters Spd and Sdv decreased but Sda increased. Similar tendency of parameter changes to milled surface analysed previously occurred; peak surface part changed more than valley portion.

Parameters of textured surfaces shown in Figure 1c changed slightly due to disturbances. The maximum change of parameters was 1% (peak density Spd and the Sdv parameter decreased). The following other parameters decreased: Spk, Sa1, Sdr, Vvc and Sda, but Sxp increased.

Changes of the parameters from polished surface shown in Figure 1d were comparatively large. The Sdv parameter decreased of about 8%, Sr1 increased of 2.4% and consequently Sa1 increased of 3.5%, Ssk decreased of 3% and Sda decreased of 1.6%. Variations of other parameters were smaller than 1%, the following parameters decreased: Sk, Sr2, Sp, Sa, Std, Sdq, Sdr, Vvc and Spd, but the following parameters increased: Spk, Sa2, Sku and Vm

Generally after analysis of parameters changes of all the tested surfaces it was found that the deviations of the skewness Ssk and kurtosis Sku occurred (up to 3%). Statistical parameters describing height Sa and Sq usually decreased,

the parameters describing total height S_z , S_p and S_v were more stable. As the results of amplitude decrease, hybrid parameters also decreased, changes of S_{dr} were higher, up to 2%. Since non-measured points were located in the peak regions of the analysed surfaces, the parameter connected with peak part S_{r1} , usually S_{pk} , sometimes S_k and consequently S_{a1} usually changed (up to 4%), in most cases peak density S_{pd} slightly decreased. Spatial parameters were usually stable, the main directionality changed only for anisotropic one-directional texture (Figure 1d). High changes of the feature parameters: S_{dv} and S_{da} deserves attention. Parameters variations depend on surface type. Maximum changes of parameters shown in the Figure 1c was 1%; it is difficult to find filled in non-measured points in this graph.

THE STUDY OF SURFACES MEASURED UNDER DIFFERENT CONDITIONS

It was found that improper measurement conditions might cause serious inaccuracies in parameters calculations. High errors of parameters computations occurred even for very small ratio of non-measured points (NMP). Figure 2 presents example. Non-measured points of small ratio 0.0268% located in valley surface portions caused increase of roughness height in this part (the S_v parameter increased by more than 20%). In addition, the spatial parameters changed; the autocorrelation length S_{al} increased by more than 20%, texture parameter S_{tr} by more than 15%. Peak density decreased by about 25%, but feature parameters: S_{da} , S_{sha} , S_{dv} and S_{shv} increased by more than 25%. These changes are large therefore study of surface distortion due to non-measuring points presence deserves attention. However changes of parameters characterizing material ratio curve, from the 3D extension of ISO 13565-2 family: S_k , S_{pk} , S_{vk} , S_{r1} and S_{r2} were smaller than 5%.

Due to growth of light intensity from the initial small value, the NMP ratio decreased, reached the smallest value and then increased.

Too small light intensity caused presence of non-measured points in valley region, but too high – in peak surface portion. Increase of light intensity caused usually growth of height, hybrid and functional (V family) parameters as well as of oil capacity, however for very high NMP ratio (larger than 40%) different tendencies of parameters changes were also possible. The spatial parameter S_{al} mainly decreased due to light intensity increase. Its changes are higher

for anisotropic surface compared to isotropic or mixed. Variability of the S_{dq} parameter is smaller than that of the S_{dr} . Similar findings was found in other research [5, 6] and in the first part of the present study.

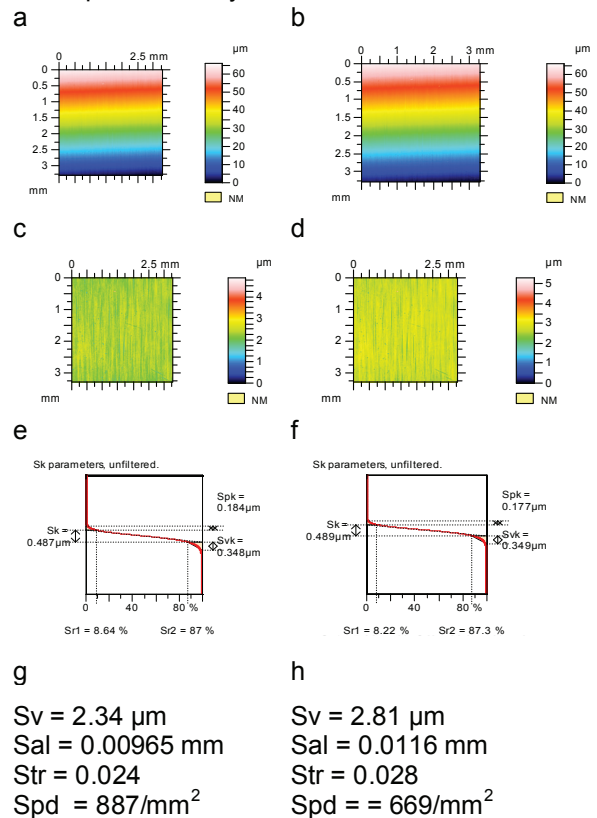


FIGURE 2. Contour plots of polished surface before (a, b) and after form removal (c, d) with material ratio curves (e, f) and selected parameters (g, h); graphs a, c, e, g correspond to NMP ratio of 0.00935% but b, d, f and h of 0.0268%.

The non-measuring points caused usually increase of the S_{sk} parameter and decrease of the S_{ku} parameter of one-process surfaces. Functional parameters S_{mc} and especially S_{xp} were more stable than the S_{mr} parameter. Big variation of the S_{mr} parameter was found in other research [5, 6]. Usually changes of spatial parameters S_{tr} and especially S_{al} were larger than those of height parameters, however different situation was also possible. The main texture direction S_{td} was constant for anisotropic surface with the main directionality, contrary to isotropic textures. It is difficult to find tendency of changes of peak density S_{pd} , however variation of this parameter was large. The following feature parameters: S_{da} , S_{dv} , S_{sha} and S_{shv}

were also non-stable on surface. Variability of the Sr2 material ratio was lower than that of Sr1 ratio. Usually one-process surface with the highest NMP ratio corresponded to maximum changes of parameters.

Generally, variability of statistical height parameters Sa, Sq, spatial parameter Std, slope Sdq, functional parameters from V family as well as material ratio Sr2 caused by non-measured points presence was relatively small. High errors existed only for high NMP ratio. However the following parameters are susceptible for errors caused by high NMP ratio: skewness Ssk, functional parameter Smr and the following feature parameters: Spd, Sda, Sdv, Sha and Shv.

Changes of parameters depend on location of non-measured points. When non-measured points are located in valley part, changes of the Sp parameter were smaller than those of the Sv parameter, contrary to non-measured points located in the peak parts of the surfaces. Figure 3 presents an example. Surface after polishing of a little asymmetric distribution is studied. The NMP ratio of surface shown in Figure 3a, b is similar to that presented in Figure 3d, e, however it is visible both from the analysis of contour plots and extracted profiles that non-measured points are located in valley and peak parts, respectively. One can see that due to non-measured points presence in valley part the Sv parameter decreased to about 40% and Sq parameter to 12%. Presence of non-measured points in peak surface part led to growth of the Sp parameter to more than 15% and decrease of the Smr parameter by more than 60%. Due to non-measured point presence the Ssk parameter increased. Change of the Sal parameter was smaller than 10%, which was probably connected with the fact that mixed surface (Str about 0.35) was analysed.

The statistical parameters Sq and Sa are stable for extremely high number of non-measured points. When these points are located in peak surface part (NMP ratio was 58% - see Figure 4c, and 4d) the value of the Sq parameter was overestimated by 9%. Similar error was obtained when non-measured points were presented in valley part (NMP ratio was 24.2% - see Figure 4a and 4b). Similar to Figure 3, presence of non-measured points in valley surface portion caused high changes of the parameter Sv, but in peak part - of the parameter Sp. In both cases, errors led to increase of the parameters Sal, Str and Ssk.

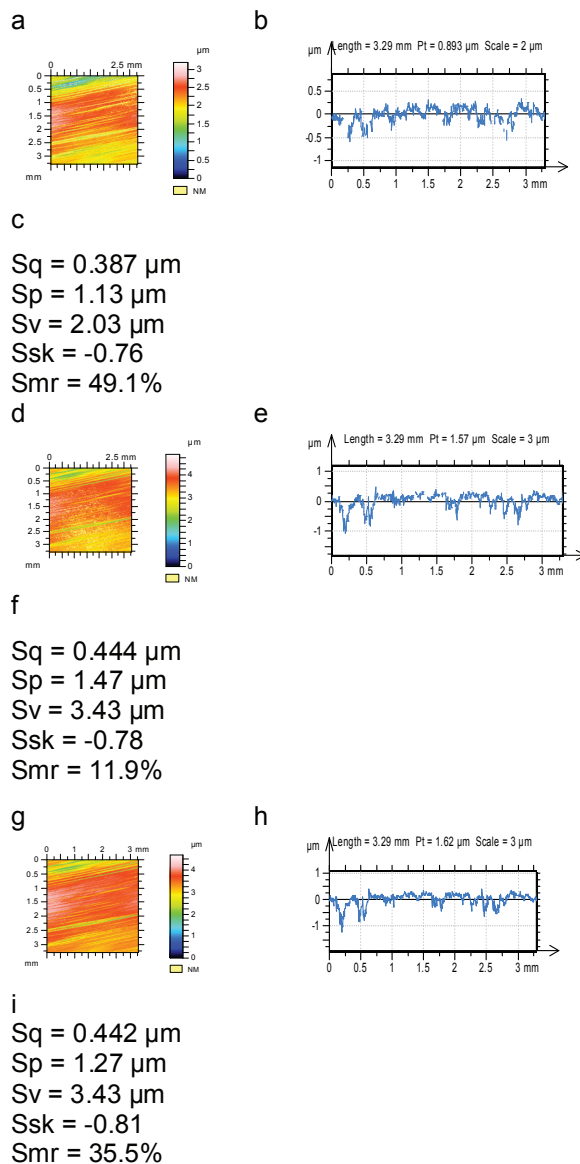


FIGURE 3. Contour plots (a, d, g), extracted profiles (b, e, h) and selected parameters (c, f, i) of polished surface with non-measured points located in valley part, NMP ratio 10.1% (a, b, c) with non-measured points located in peak part, NMP ratio 9.3% (d, e, f) with the smallest number of non-measured points, NMP ratio 0.1% (g, h, i).

The presented above tendencies are concerned mainly for one-process surfaces. However similar trends were observed also for two-process textures of negative skewness (usually smaller than -1). Increase of light intensity led in most cases to decrease of skewness Ssk and

usually to increase of kurtosis S_{ku} . This is reasonable because small values of skewness correspond to large values of kurtosis. This tendency was probably caused by presence of non-measured points in the bottom of the valleys (see Figure 3a and 3b).

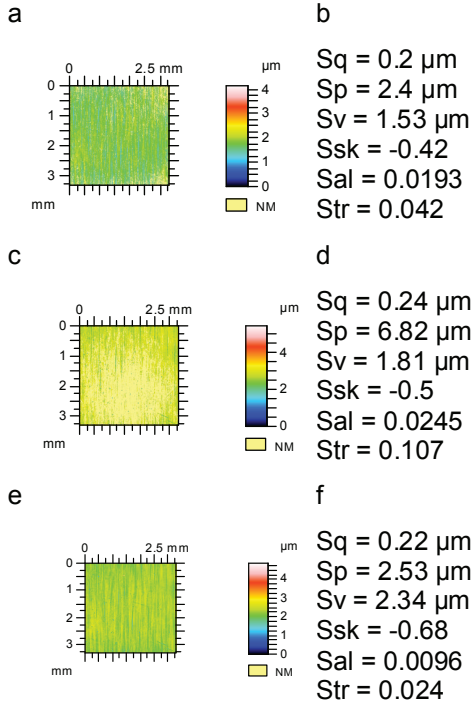


FIGURE 4. Contour plots (a, c, e) and selected parameters (b, d, f) of polished surface with non-measured points located in valley part, NMP ratio 24.2% (a, b) with non-measured points located in peak part, NMP ratio 58% (c, d) with the smallest number of non-measured points, NMP ratio 0.00935% (e, f).

In surface shown in Figure 3 change of the S_{sk} parameter is small, but high errors of this parameter calculation can occur for two-process textures. As a result surface machined well looks like one-process texture and can be classified as spoilage. Increase of the S_{sk} parameter is connected with decrease of the S_{vk} parameter. Errors of two-process surface topography parameters estimations are very important, since functional properties of two-process surfaces are better than those of one-process textures [7-9]. Inaccuracy of measurement of valley part usually led to larger errors of parameters computations compared to disturbances in plateau region. For instance the relative errors of parameters of surface shown in Figures 5a and 5b are higher than those of surface shown in Figures 5c and 5d although the NMP ratio of the

latter surface is higher. The parameters S_q , S_p , S_z and S_v of surface shown in Figures 5a and 5b were underestimated by 19%, 32%, 44% and 48%, respectively. However these parameters of surface shown in Figures 5c and 5d were overestimated by 11%, 11%, 10% and 9%, respectively. It is maybe more important that the S_{sk} parameter of surface shown in Figure 5a and 5b is -0.34, 5c and 5d -1.26 but 5e and 5f -1.25.

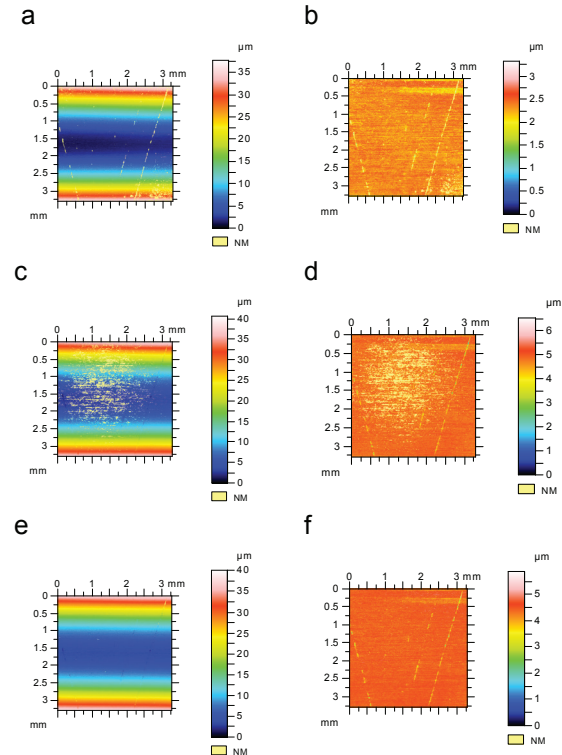


FIGURE 5. Contour plots of worn cylinder surface before (a, c, e) and after form removal (b, d, f) with non-measured points located in valley region; NMP ratio 4.31% (a, b), in peak part; NMP ratio 14.4% (c, d) and of their smallest number; NMP ratio 0.102% (e, f).

The other tendencies of parameter changes due to non-measured points presence found for one-process surface were confirmed for two-process textures. The texture direction was mainly constant. The features parameters, especially S_{da} , S_{ha} , S_{dv} and S_{hv} were non-stable on two-process surfaces.

Non-correct light intensity could lead to presence of spikes (see Figure 6). However non-measuring point erasing for the whole surface can cause increase of profile height (Figure 6c, 6e), opposing to similar correction of only profile (Figure 6g). When light intensity was smaller

(Figure 6b, 6d, 6f and 6h) the non-measured points appeared in the same place (see arrows) but without presence of spikes. The same profile was obtained after correction of the whole surface or profile (see Figure 6f and 6h). The Sp parameter of corrected surface of smaller NMP ratio was $1.21\text{ }\mu\text{m}$, but of higher $3.07\text{ }\mu\text{m}$.

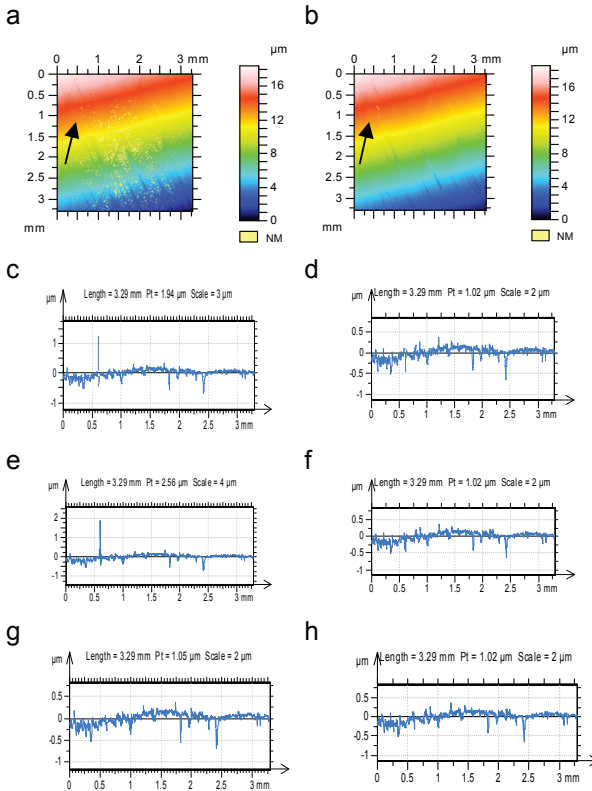


FIGURE 6. Contour plots (a, b), extracted profiles before correction - filling in (c, d), after correction of the whole surfaces (e, f) and after profile correction (g, h), NMP ratio 6.68% (a, c, e, g) and 0.079% (b, d, f, h).

CONCLUSIONS

Presence of even small number of non-measured points can cause false estimation of parameters.

When the non-measured points are located in one place in peak surface part, parameters describing this part changed more than the others. Character of parameters changes depends on the surface type.

Increase of light intensity caused usually increase of height, hybrid and functional parameters; the spatial parameter Sal mainly decreased. The following parameters are susceptible for errors caused by high NMP ratio: skewness Ssk, Smr and feature parameters: Spd, Sda, Sdv, Sha and Shv.

For stratified surfaces increase of light intensity caused decrease of skewness Ssk and increase of kurtosis Sku. These changes can be large. For this type of surface inaccuracy of measurement in valley part led usually to larger errors of parameters computations compared to disturbances in plateau region.

REFERENCES

- [1] Pawlus P, Wiczorowski M, Mathia T. The errors of stylus methods in surface topography measurements. Zapol, Szczecin 2014.
- [2] Whitehouse DJ. Surface metrology today: complicated, confusing effective? Proceedings of the 13th International Conference on Metrology and properties of Engineering Surfaces. Twickenham Stadium (UK); 2011: 1-10.
- [3] de Groot P. Coherence scanning interferometry. Chapter 9 in Optical Measurement of Surface Topography. R. Leach (editor), Springer-Verlag, Berlin-Heidelberg; 2011.
- [4] Podulka P, Pawlus P, Dobrzański P, Lenart A. Spikes removal in surface measurement. Journal of Physics: Conference Series. 2014; 483(1): 012025.
- [5] Pawlus P, Grabon W, Reizer R, Gorka S. A study of variations of areal parameters on machined surfaces. Surface Topography: Metrology and Properties. 2015; 3(2): 025003.
- [6] Dzierwa A, Reizer R, Pawlus P, Grabon W. Variability of areal surface topography parameters due to the change in surface orientation to measurement direction. Scanning. 2014; 36(1): 170-183.
- [7] Grabon W, Pawlus P, Sep J. Tribological characteristics of one-process and two-process cylinder liner honed surfaces under reciprocating sliding conditions. Tribology International. 2010; 43(10): 1882-92.
- [8] Dzierwa A, Pawlus P, Zelasko W. Comparison of tribological behaviors of one-process and two-process steel surfaces in ball-on-disc tests. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology. 2014; 228: 1195-1210.
- [9] Pawlus P. Effects of honed cylinder surface topography on the wear of piston-piston ring-cylinder assemblies under artificially increased dustiness conditions. Tribology International. 1993; 26/1: 49-55.

MORPHOLOGICAL FILTRATION OF STRATIFIED TEXTURES

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INTRODUCTION

Filtration techniques are widely used as the pre-processing tools to remove unwanted components of surface texture. Mean-line based filters are commonly used. Morphological filters, being their complement are believed to give better results for stratified surface topography. They have found many applications in practice. A direct algorithm of traditional envelope filter was developed by Shunmugam and Radhakrishnan [1]. Morphological filters are constructed using structural elements. Usually the circles and horizontal segments are considered [2]. The two basic morphological operations are dilation and erosion. During a dilation, the structuring element is in contact with and above the surface, but during an erosion, the structuring element is in contact with and below the texture. The closing filter is the result of a dilation followed by an erosion. The closing filter defines the upper envelope of the surface, although the traditional envelope filter [2] was equivalent to the dilation operation. The opening filter is the result of an erosion followed by a dilation, this filter defines the lower envelope. It is possible to apply a closing filter followed by an opening filter, or in the reverse way, an opening filter followed by a closing filter; these filters are called alternating sequence filters. Morphological filters are believed to offer more possibilities than traditional mean-line filters, such as multi-scale analysis [2], eliminating high peaks or valleys, reconstructing mechanical surface after stylus measurement [3] and simulating the conformable interface of two mating surfaces [4]. They can be also used in contact analysis [3]. Algorithms for morphological profile filters were presented in Reference [5].

Two-process (stratified) texture is more important from functional point of view than one-

process surface. Plateau honed cylinder surface is the typical example of two-process texture. It consists of smooth plateaus with intersecting deep valleys. It combines good sliding properties of a smooth surface and great ability to maintain oil of a porous texture. It consists smooth wear-resistant and load-bearing plateaus and deep valleys working as oil reservoirs and debris trap. Plateaued surfaces have been machined to simulate those resulting from normal running-in. Plateaued surfaces have advantage over one-process textures [6, 7, 8, 9, 10]. Plateau honed surfaces are kind of textured surfaces, of very good tribological properties [7].

One of the application of morphological filters is the analysis of two-process textures. The authors of References [3, 11] confirmed usefulness of these filters for plateau-honed cylinder surface.

SELECTION OF SPHERE RADIUS

The present authors analysed morphological filtration of stratified textures. Simulated profiles contained plateaus of a given roughness height, characterized by its standard deviation Ppq , and triangular scratches of various sizes as well as plateau-honed cylinder profiles and other textured surfaces were studied. Dilation, closing and sequence (closing + opening) filters were used with circle (disk) as structuring element. There is problem of determining the correct value of its radius. It is related to depth of penetration of circle inside the valley h of width a (see Figure 1):

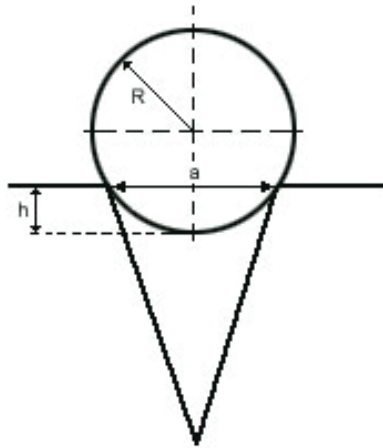


FIGURE 1. Scheme of estimation of disk radius R of morphological filter.

$$h = R - \sqrt{R^2 - 0.25a^2} \quad (1)$$

Therefore disk radius should be:

$$R = \frac{h^2 + 0.25a^2}{2h} \quad (2)$$

The depth h was determined on the basis of simulated profiles filtering. It was assumed that it should be small fraction of waviness total height. Since waviness amplitude is usually higher for bigger texture height, the depth h is higher when the Ppq parameter is higher. The following formula was used for h estimation:

$$h = -0.158 Ppq^2 + 0.355 Ppq \quad (3)$$

Profiles with the Ppq parameter smaller than $1 \mu\text{m}$ were analysed in estimation of the depth h . The Ppq parameter is equal to the standard deviation of surface height of plateau surface. This parameter can be easily calculated when plateau and valley parts are of random nature (see ISO 13565-3 standard). There is problem of computing it for random-deterministic textures, when valley part is of deterministic character (surfaces with isolated oil pockets is the example of those textures). However Grabon et al. presented method of the Ppq parameter computation for textured surfaces with a deterministic pattern of valley features [12].

The proposed procedure of disk radius selection could be easily extended into areal (3D) surface topography analysis.

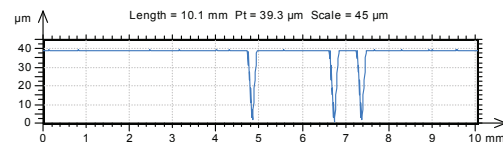
THE ANALYSIS OF SIMULATED PROFILES

Behaviors of morphological filters were compared with those of double Gaussian (Rk)

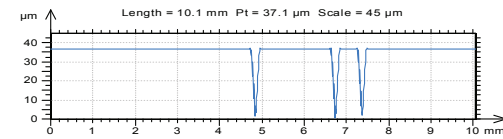
filter. Robust filter was also taken into consideration. In calculation, TalyMap software was used.

Since plateaus of modeled profiles having triangular scratches did not contain substantial waviness, filtered and unfiltered parameters were compared. It was found after analysis of simulated profiles that morphological filters reacted differently to the Rk filter on wide and deep valleys. After morphological filter usage (especially closing or dilation filter) the emptiness coefficient Rp/Rt (Rp - peak height, Rt - total roughness height) decreases (after alternating sequential filter application the emptiness coefficient sometimes, for smooth plateau surface, slightly increased).

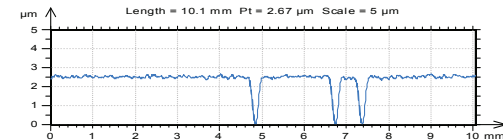
a



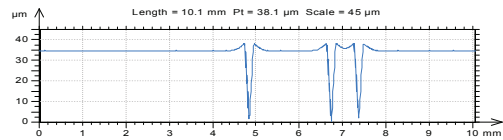
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d



e

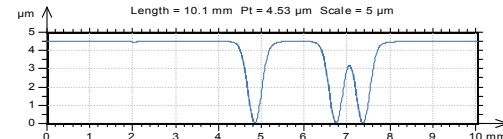


FIGURE 2. Simulated unfiltered profile A – $Pp = 1.41 \mu\text{m}$ (a), roughness profile filtered by dilation filter of 5 mm diameter - $Rp = 1.16 \mu\text{m}$ (b), waviness profile filtered by dilation filter of 5 mm radius (c), roughness profile filtered by double Gaussian filter of 0.8 mm cut-off - $Rp = 4.18 \mu\text{m}$ (d), waviness profile filtered by double Gaussian filter of 0.8 mm cut-off (e).

However, after double Gaussian filter application the Rp/Rt ratio always increased.

The differences between filters behaviors were due to the fact that morphological filters caused change of profile inside the valleys, but the double Gaussian filter (similar to Gaussian filter) affects also neighboring points to the valleys. This is evident in Figure 2 presenting simulated profiles with three valleys of width 200 μm and depth 40 μm , Ppq parameter was 0.1 μm . Application of traditional envelope filter (dilation) of too small diameter of 5 mm caused decrease of emptiness coefficient from 0.036 to 0.032, but use of the Rk filter (cut-off 0.8 mm) led to its increase to 0.11. After application of the Rk filter overshoot appeared.

The other important difference is that the depth of valley is always irrelevant for morphological filters use; this filter reacts mainly on the valley width. Contrary, not only valley width but also depth affect behavior of the Rk filter; roughness distortion due to improper filter application is bigger for higher valley depth. Depth of valleys shown in Figure 3a, of 200 μm width (Ppq parameter was 0.1 μm), was reduced about 5 times (Figure 3b). However, waviness profiles after using alternative sequence filter of these two profiles are identical – shown in Figure 3c. Contrary, maximum height of waviness profile filtered by Rk filter depends on the valley depth – is about 3 times smaller after filtering profile with reduced valley depth. In order to shown changes of filtered profiles disk of morphological filter was too small.

Many simulated profiles of different standard deviations of plateaus, number of triangular valleys and their sizes were tested. It was found that from among morphological filters, sequence filter application caused the smallest, but closing – the highest distortion of the roughness profile. After closing filter application, the Rpq parameter considerably decreased and change of plateau part occurred; it is interesting that the highest points of peaks were on the same vertical level, resulting in considerable decrease of the Rpk parameter, sometimes to 0 value and decrease of the Rp parameter. This behavior was repeated for various profiles. As the results, the profiles were distorted. Please note that the Rpq parameter is within the requirements of the leading engine builders with regard to plateau-honed cylinder profiles.

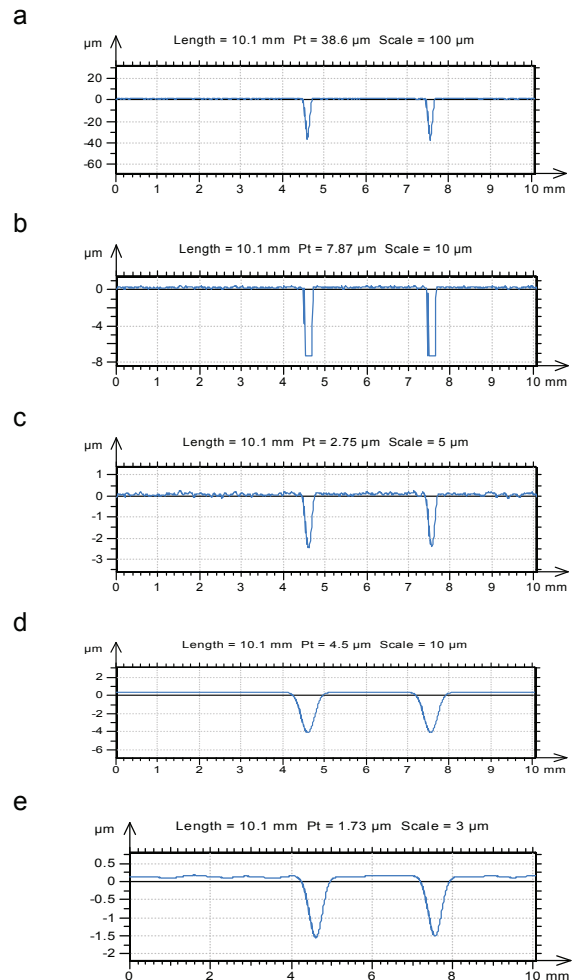


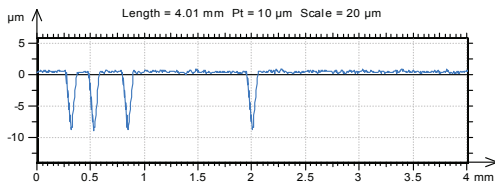
FIGURE 3. Simulated unfiltered profile B (a), profile B with reduced valleys depths B1 (b), waviness profile B and B1 filtered by dilation filter of 5 mm diameter (c), waviness profile B filtered by double Gaussian filter of 0.8 mm cut-off (d), waviness profile B1 filtered by double Gaussian filter of 0.8 mm cut-off (e).

Figure 4 presents profile characterized by the Ppq parameter of 0.2 μm having four valleys of width 100 μm and depth about 10 μm . This profile was filtered using closing filter of 40 mm radius. It is evident from zoomed detail of the roughness profile that the highest point of all the peaks are located on the same level. Consequently the Rpk parameter decreased from 0,161 μm value to 0, but the Ppq parameter decrease to 0.14 μm .

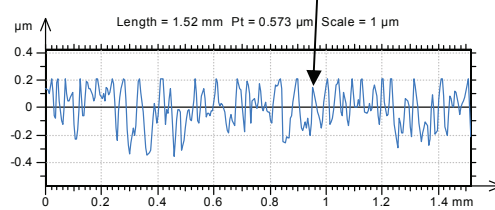
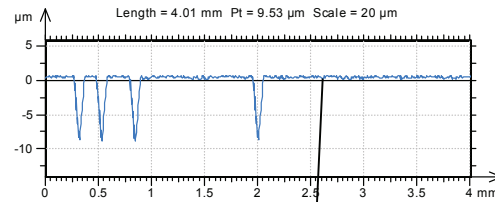
Increase of cut-off from 0.8 mm to 2.5 mm caused decrease of profile distortion, when the Rk filter was used. Due to the robust filter

application distortion were usually small, except to case when several deep valleys were close to each other. Robust Gaussian filter reacts similarly to the Rk filter on deep valley presence.

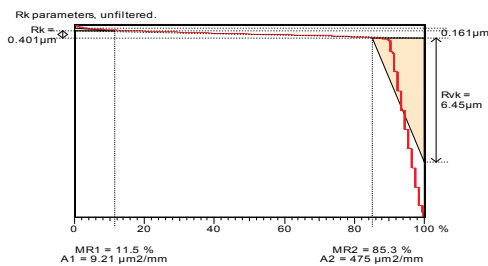
a



b



c



d

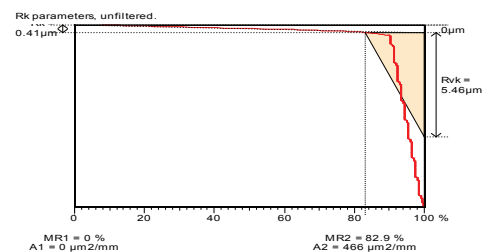
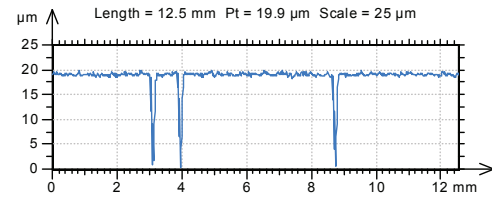


FIGURE 4. Simulated unfiltered profile C (a), roughness profile filtered by closing filter of 40 mm diameter and enlarged detail (b), material ratio of unfiltered profile (c) and material ratio of roughness profile (d).

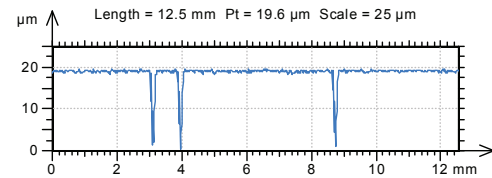
Figure 5 presents example of simulated profile D. The Pp parameter was 0.3 μ m, valleys widths and depths were about 160 μ m and 20 μ m, respectively. One can see that application of

sequential filter caused small change of the shape of roughness profile. The emptiness coefficient Rp/Rt decreased from 0.059 to 0.052. However, increase of the diameter of disk from 60 mm to 125 mm (the largest possible value) didn't change the emptiness coefficient. For comparatively rough plateau surface application of morphological filter with disk as structuring element caused decrease of the emptiness coefficient because edges of the deep valley are located under the profile highest peaks.

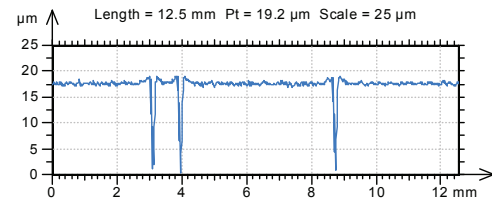
a



b



c



d

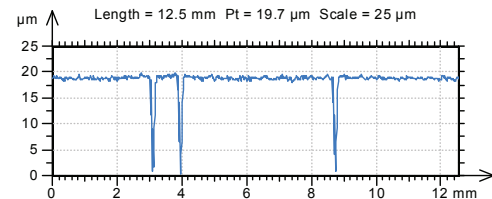


FIGURE 5. Simulated unfiltered profile D – $Pp = 1.18 \mu$ m, $Pt = 9.6 \mu$ m (a), filtered by alternating sequence filter of 60 mm diameter – $Rp = 1.02 \mu$ m (b), filtered by double Gaussian filter of 0.8 mm cut-off – $Rp = 1.8 \mu$ m (c) and filtered by double Gaussian filter of 2.5 mm cut-off – $Rp = 1.29 \mu$ m (d).

Application of the Rk filter with 0.8 mm cut-off caused profile distortion (see Figure 5c) and increase of emptiness coefficient to 0.094 value. Growth of cut-off to 2.5 mm led to smaller increase of the Rp/Rt ratio – to 0.065. The

spacing parameter RSm decreased from 2.81 mm to 1.87 mm after application of double Gaussian filter with cut-off of 2.5 mm as well as of morphological filter and to 0.94 mm after use of the Rk filter with cut-off of 0.8 mm.

The form component needs to be removed from the profile before the Rk or robust filters could apply to the data. However, it was found after analysis of many simulated profiles with added curvatures that using closing filter, the profile didn't need to be preprocessed to form removal. This is important because not correct form removal can lead to distortion of stratified textures [12].

THE ANALYSIS OF MEASURED PROFILES

Similar observations were obtained on the basis of measured textures. For measured two-process surfaces, like cylinder liner texture after plateau honing, estimation of sizes of grooves seems to be difficult. However, widths of valleys can be determined using procedure describe in Reference [13].

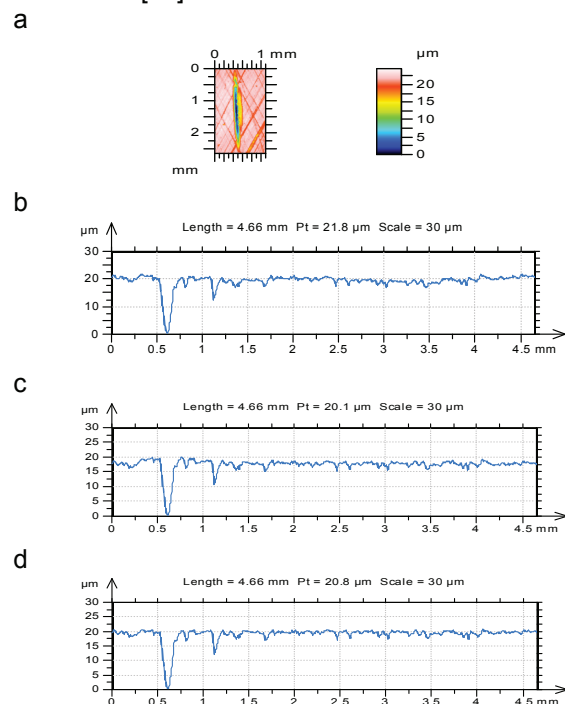


FIGURE 6. Detail of contour map of cylinder profile with isolated oil pocket (a), extracted profile E containing dimple, $P_p = 2.51 \mu\text{m}$, $PSm = 0.244 \text{ mm}$ (b), filtered by double Gaussian filter of 0.8 mm cut-off - $R_p = 2.72 \mu\text{m}$, $RSm = 0.194 \text{ mm}$ (c) and filtered by alternating sequence filter of 46 mm diameter $R_p = 2 \mu\text{m}$, $RSm = 0.221 \text{ mm}$ (d).

Generally the proposed procedure of disk radius of morphological filter selection was found to be very useful also for measured surfaces. It can be also used for profile with isolated dimples. Figure 6 presents example of profile containing separate long oil pocket.

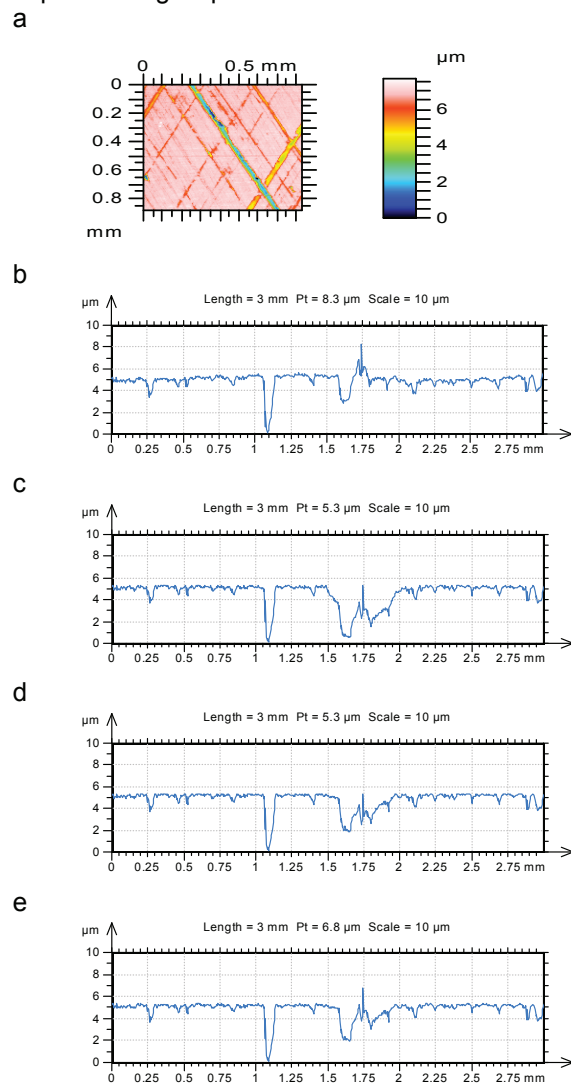


FIGURE 7. Detail of contour map of plateau honed cylinder profile (a), extracted profile F containing spike (b), filtered by dilation filter of 22 mm diameter (c), filtered by closing filter of 22 mm diameter (d) and filtered alternating sequence filter of 22 mm diameter (f).

After using the Rk filter the emptiness coefficient increased from 0.115 to 0.135, but after application of sequential filter it decreased to 0.096. Decrease of the RSm parameter after morphological filter application was lower than after double Gaussian filter usage. One can see

that shape of deep valley changed after the Rk filter application contrary to morphological filter use.

Unfortunately morphological filters are very sensitive on spikes on the profile presence [2]. Near this spike usually depression is formed. Distortions were the smallest after using the sequential filter from among morphological filters – see Figure 7. Outliers should be eliminated before morphological filter application. Use of double Gaussian or robust Gaussian filter did not lead to profile distortion.

CONCLUSIONS

The special procedure of disk radius selection was elaborated in order to filter two-process textures. The proposed procedure was found to be very useful for simulated and measured surfaces. It was developed for 2D profiles but could be easily extended for areal (3D) surface topography filtering. Behaviors of various morphological filter were compared with that of valley suppression filter. From among morphological filters, the alternate sequential filter is recommended. Closing filter is the best for form removal. Morphological filters are very sensitive on spikes on the profile presence.

REFERENCES

- [1] Shunmugam MS, Radhakrishnan V. Two- and three dimensional analyses of surfaces according to the E-system. *Proceedings of the Institution of Mechanical Engineers*. 1974; 188: 691-699.
- [2] Scott PJ. Scale-space techniques. *X International Colloquium on Surfaces*. Chemnitz 2000; 153-161.
- [3] Lou S, Jiang X, Scott PJ. Application of morphological operations in surface metrology and dimensional metrology. *Proceedings of the 14th International Conference on Metrology and Properties of Engineering Surfaces*. Taipei 2013.
- [4] Malburg CM. Surface profile analysis for conformable surfaces. *Transactions of The ASME, Journal of Manufacturing Science and Engineering*. 2003; 125: 624-627.
- [5] Lou S, Jiang X, Scott PJ. Algorithms for morphological profile filters and their comparison. *Precision Engineering*. 2012; 36: 414-423.
- [6] Jeng Y. Impact of plateaued surfaces on tribological performance. *Tribology Transactions*. 1996; 39/2:354-61.
- [7] Grabon W, Pawlus P, Sep J. Tribological characteristics of one-process and two-process cylinder liner honed surfaces under reciprocating sliding conditions. *Tribology International*. 2010; 43(10): 1882-92
- [8] Dzierwa A, Pawlus P, Zelasko W. Comparison of tribological behaviors of one-process and two-process steel surfaces in ball-on-disc tests. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*. 2014; 228: 1195-1210.
- [9] Pawlus P. Effects of honed cylinder surface topography on the wear of piston-piston ring-cylinder assemblies under artificially increased dustiness conditions. *Tribology International*. 1993; 26/1: 49-55.
- [10] Etsion I. State of the art in laser surface texturing. *Proceedings of the 12th Conference on Metrology and Properties of Engineering Surfaces*. Rzeszow (Poland) 2009: 17-20.
- [11] Muralikrishnan B, Raja J. Functional filtering and performance correlation of plateau honed surface profiles. *Manufacture science and Engineering*. 2005; 127: 193-197.
- [12] Grabon W, Pawlus P, Galda L, Dzierwa A, Podulka P. Problems of surface topography with oil pockets analysis. *J. Phys. Conf. Ser.* 2011; 311: 012023.
- [13] Pawlus P, Reizer R, Wieczorowski M. The analysis of directionality of honed cylinder liners surfaces. *Scanning*. 2014; 36(1): 95-104.
- [14] Podulka P, Dobrzański P, Pawlus P, Lenart A. The effect of reference plane on values of areal surface topography parameters from cylindrical elements. *Metrology and Measurement Systems*. 2014; 21(2): 247-256.

OPTIMAL SPECIMEN ORIENTATION IN CONE-BEAM X-RAY CT SYSTEMS (FOR DIMENSIONAL METROLOGY)

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X-ray computed tomography (CT) has attracted the interest of major companies in automotive, electronics, aerospace, medical devices, cast materials, plastics, rubber, 3D printing, and leisure industries as well as many other industrial fields. Given its potential as a precise, non-contact measuring technique, research centers are currently investing in improving and expanding CT systems' capabilities for industry metrology applications. This paper presents the results of an experimental study conducted at the University of North Carolina Charlotte (UNCC) to investigate the effects of specimen orientation/position in industrial CT metrology for some typical 3D shapes: a rectangular block, a hollow rectangular block, a cylinder, and a hollow cylinder (see Figure 1). The specimens were manufactured in aluminum as well as in thermoplastic PMMA (Poly(methyl methacrylate)), and had dimensions 60 mm long with major dimensions of 20 mm X 40 mm for the blocks, 20 mm radius for the cylinder, and 10 mm radius for the holes. It was found that the angular positions to obtain the smallest deviations between dimensional measurements using contact CMM and CT scans for the samples studied here lie in a range of 10 to 40 degrees, see Figure 2. It was also observed that angular positions above 60 degrees strongly affect the volume reconstruction of the parts, sometimes even creating non-existing features (artificial artifacts), and hence present considerable problems with an assessment of the accuracy of CT dimensional measurements.

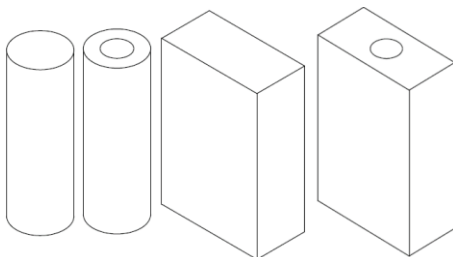


FIGURE 1. Geometrical shapes for the cylinder, hollow cylinder, block, and hollow block.

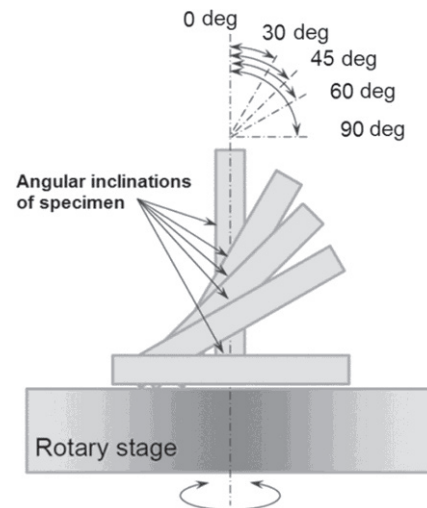


FIGURE 2. Inclined orientations of specimen at different angular positions in a CT rotatory table.

BACKGROUND OF THE RESEARCH

Although CT technology holds great potential for measuring industrial parts, there are many aspects of its accuracy and traceability that still need to be understood [1-4]. A major unresolved problem is the presence of CT artifacts (i.e., artificial structures in the scan results). The major cause of CT artifacts are physical effects like beam hardening and scattering radiation, but also with cone-beam x-ray sources, Feldkamp artifacts (blurriness in the resulting reconstructed volume) can be generated by the reconstruction algorithm itself (called the FDK: Feldkamp-Davies-Kress algorithm) [5]. This latter effect occurs because, for some of the planar faces of the specimen, an exact reconstruction is limited due to the incompleteness of the acquisition geometry using a circular cone-beam system [6, 7]. For example, in Figure 3, a missing cone of frequencies can be observed in an image generated by a cone-beam X-ray CT system that used a Shepp-Logan noise suppression filter. This object comprised a stack of equally spaced disks that should show strong Fourier peaks along the vertical axis of the measured 2-D FFT.

When using an inclined position for the sample within the CT scanner cavity at $Z=0$, the missing cone of frequencies in the Fourier domain appear and the reconstruction method will have the information necessary to perform a more complete reconstruction of the object in the spatial domain.

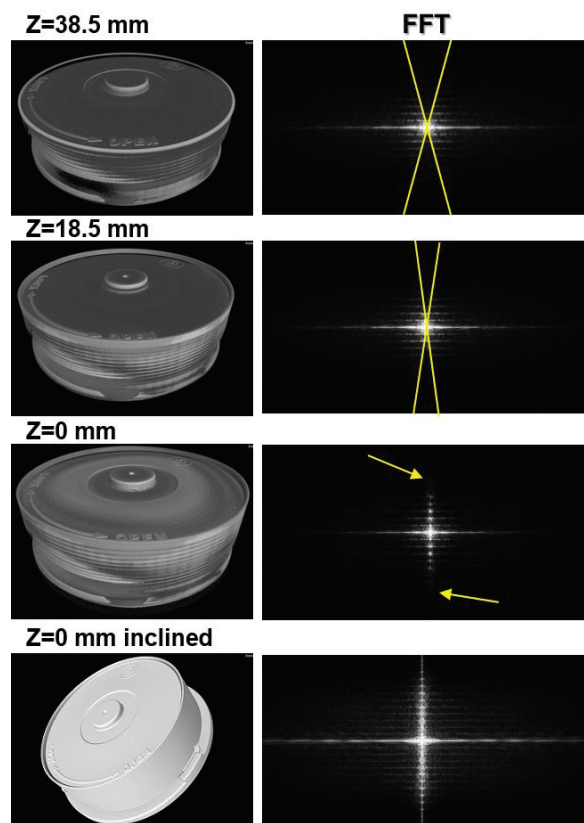


FIGURE 3. *Experimental CT images for a stack of DVD's. Left: Feldkamp artifacts present in the CT reconstruction when varying its position along the principal axis (Z-axis) of the rotatory table. Right: Fourier domain for CT slices parallel to the Z-axis and cutting through the center of the DVD stack. $Z=0$ corresponds to a point on the line joining the X-ray point source with the center of the detector and sitting at the center of the object.*

Feldkamp artifacts cannot be corrected since they are produced by the absence of information in the reconstruction algorithm itself. The presence of other CT artifacts in the reconstructed image resulting from known physical effects depends on the accuracy of selection settings in the machine. These settings modify the physical characteristics of the X-ray beam and the detection system and require consideration of geometry, material, and position

and orientation of the specimen inside the scanner. When the CT artifacts caused by physical effects are not very pronounced, errors during the reconstruction (mainly beam hardening and scattering) can be resolved using software correction algorithms. With further research, the accuracy of the measurements can be improved not only by optimizing the scanning settings for the CT machine and the reconstruction software but also by avoiding or minimizing the presence of Feldkamp artifacts in the reconstructed 3D image. One way this can be achieved is by selecting, for example, an appropriate orientation/position of the specimen inside of the scanner, which means optimizing the specimen placement inside the cone-beam CT configuration. The optimization of specimen orientation for metrology purposes when using cone-beam X-ray CT systems forms the focus of the research presented in this paper.

THE HOLLOW BLOCK

Due to limitations of space, analysis of CT results presented in this paper is restricted to the aluminum and PMMA hollow blocks only. The analysis of CT measurement results for the other geometrical shapes introduced in Figure 1 are presented in reference [8]. The mean error of CT measurements (respect to CMMs) under variations of axial tilt of the workpiece during the CT scan are shown in Figures 4 through to Figure 8. Each figure presents the mean error of the CT measurements (of the same type) taken at five different locations on the workpiece (shown in the sketches included on the figures), with respective standard deviations from the five measurements represented by vertical bars. The measurement values are the average results of three separate measurements coming from three scan repetitions. Figure 4, Figure 5, and Figure 6 show deviations from CMM values for CT measurements of size (width, length, and diameter), while Figure 7 and Figure 8 show respective deviations for measurements of form (roundness and flatness).

The setting parameters for scanning were: $40\text{ }\mu\text{m}$ focal spot size, $54\text{ }\mu\text{m}$ voxel size, 2.33 geometrical magnification, 330 mm source-object distance, 800 mm source-detector distance, 800 number of projections, 1X1 binning mode, 5 images averaging, offset image correction with 10 images for averaging, 10x gain, gain image correction with 7 homogenization steps and 10 images for averaging, Shepp-Logan noise suppression filter, and X-ray intensity correction

in dynamic gray scale value determination mode for each image. To get enough contrast (high signal-to-noise ratio), the aluminum workpiece was scanned with X-rays generated with a power of 39 W (130 kV voltage and 300 μ A current) and pre-filtered through 1 mm thickness copper, and images were captured at an integration time rate of 400 ms (2000 ms for 5 images averaged). On the other hand, for the PMMA case, the hollow block was scanned with X-rays generated with a power of 16 W (100 kV voltage and 161 μ A current) and pre-filtered with 0.5 mm thick aluminum, and images were captured at an integration time rate of 267 ms (1335 ms for 5 images averaged). With these setting parameters for each workpiece, two different orientations O1 and O2 (see sketch on each figure, from Figure 4 to Figure 8) were chosen for CT scanning, with allowed variations on the axial tilt of the hollow block with respect to the axis of rotation inside the CT scanner. Other scanning parameters remained the same in each case.

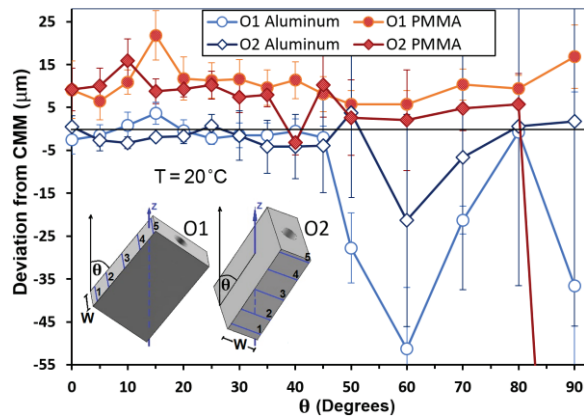


FIGURE 4. Mean error of CT measurements of width (W) for the hollow block under variations of axial tilt during scan in cone-beam X-ray CT.

Because the main purpose of this research was to find the angular positions of tilt for the workpieces that yield dimensional measurements with the smaller deviations respect to measurements taken with contact CMMs, the strategy of measurement in both techniques (X-ray CT and tactile CMM) was the same for each workpiece. This was achieved by using the same software program, Calypso 5.6 [9], with CT and CMM hardware from the same manufacturer. A Zeiss Metrotom 800 CT system [10, 11] and a Zeiss Micura CMM [12] were used for the measurements in this research. A detailed description of the measurement strategy can be

found in reference [8], but sketches indicating the locations and features in the workpiece used for measurement are included in the figures that present measurement results in this paper. The dimensional measurements coming from CT scanning of the workpieces were obtained at the standard reference $T = 20\text{ }^{\circ}\text{C}$ [13].

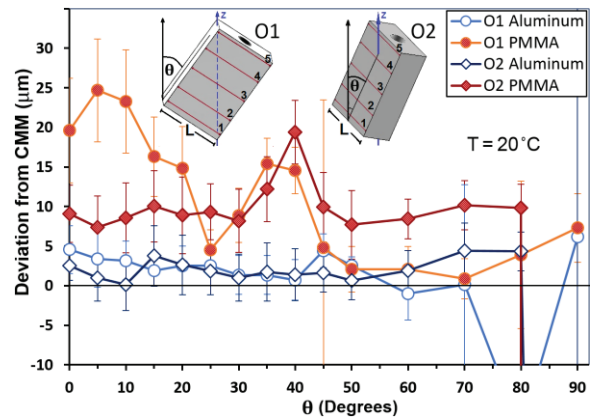


FIGURE 5. Mean error of CT measurements of length (L) for the hollow block under variations of axial tilt during scan in cone-beam X-ray CT.

From Figure 4, measurement errors (respect to CMM), using CT scanning, were obtained for the aluminum hollow block's width by scanning the workpiece at angular (axial tilt) positions between 0 to 90 degrees. Deviations less than 5 μ m were detected in the region of angular inclinations between 0 to 45 degrees for both, the orientations O1 and O2 (refer to sketch on Figure 1), with a minimal absolute error 0.4 μ m at $\theta = 20^{\circ}$ in orientation O1. Repeatability of the observations is represented by the standard deviations (vertical bars) associated to each mean value, which is 3 μ m or less at angular positions between 5 to 45 degrees for the orientation O1, and less than 3 μ m only for the range 0 to 25 degrees with the orientation O2. The standard deviations increase outside of these angular positions, particularly for the range 50 to 90 degrees, at which point the mean error of CT measurements goes up to 51 μ m for the orientation O1 and 21 μ m for the orientation O2. A similar situation is observed for measurements of length on the aluminum hollow block, see Figure 5, but for a larger range of angular positions, from 0 to 60 degrees, for which the mean errors of CT measurements are less than 5 μ m with standard deviations also less than 5 μ m.

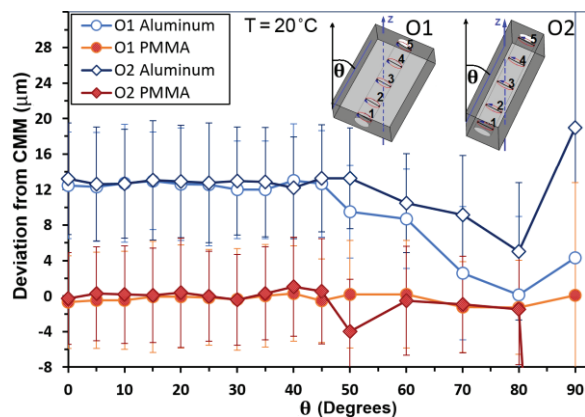


FIGURE 6. Mean error of CT measurements of diameter of the holes for the hollow block under variations of axial tilt during scan in cone-beam X-ray CT.

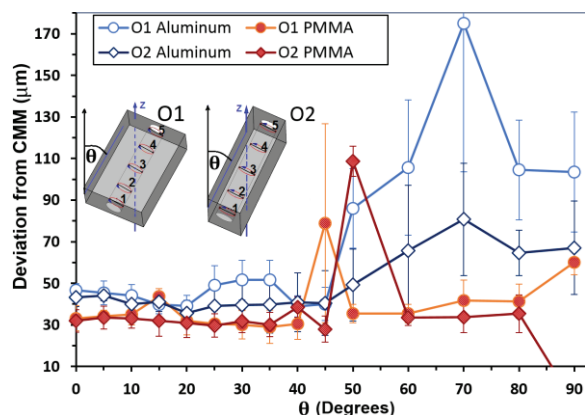


FIGURE 7. Mean error of CT measurements of roundness of the holes for the hollow block under variations of axial tilt during scan in cone-beam X-ray CT.

The smallest mean error of CT measurement of length for the aluminum hollow block was obtained for $\theta = 20^\circ$ at the orientation O2, with a deviation from CMM of $0.1 \mu\text{m}$. On the other hand, the mean errors of CT measurements of width and length on the PMMA hollow block, see Figure 4 and Figure 5, are larger; around $10 \mu\text{m}$ for most of the angular inclinations, but still with standard deviations around $5 \mu\text{m}$ or less. Hence, there seems to exist a systematic error of about 5 to $10 \mu\text{m}$ for most of the CT measurements of length (and width) compared to CMM reference measurements (except for a few angular positions between 50 and 70 degrees for length measurements at the orientation O1 and width measurements at the orientation O2). These larger length measurement errors on the PMMA

workpiece, compared to those for the aluminum piece, might come from temperature effects given the large coefficient of thermal expansion for PMMA, which is equal to about $126 \mu\text{m}\cdot\text{m}^{-1}\cdot^\circ\text{C}^{-1}$. The CT measurements for the PMMA were taken at an average temperature of $21.9 \pm 0.3^\circ\text{C}$. For the aluminium workpiece with coefficient of thermal expansion of $23.4 \mu\text{m}\cdot\text{m}^{-1}\cdot^\circ\text{C}^{-1}$, the average temperature during CT scanning was $22.3 \pm 0.3^\circ\text{C}$. The necessary corrections for temperature in the dimensional measurements were applied ($-9.6 \mu\text{m}$ for 40 mm length and $-4.8 \mu\text{m}$ for 20 mm width in the PMMA workpiece) for reporting the CT measurements to the standard reference $T = 20^\circ\text{C}$. However, it seems likely that these compensations could have been underestimated for the PMMA workpiece's length (and width) measurements, as can be inferred from Figure 4 and Figure 5. Such corrections are often complicated by thermal gradients and lack of precise thermal expansion data, particularly for the temperature inside of a CT cavity, which raises spatial and temporal temperature gradients driven by heat generated by the X-ray source [14] (and potentially other sources such as motion stages, cameras, etc.). Nevertheless, it is interesting to see that despite the assumptions of additional unknown thermal expansions, this effect seems not to be present for the hole's diameter measurements in the same workpiece, the PMMA hollow block, with the same scan data used for length and width measurements, see Figure 6. But in the same figure, there seems to exist additional systematic errors in the measurements of hole's diameter for the aluminum workpiece, for which dimensional errors from temperature effects would not be expected to be larger than one micrometer for a nominal 10 mm diameter (measured at $22.3 \pm 0.3^\circ\text{C}$). This is rather interesting and could be the effect of either beam hardening from the X-rays penetrating metallic materials or errors in material dependent threshold determination for edge detection [15-19], particularly here for dimensional metrology of internal cavities. It is also interesting to see that these effects seem to be minimized by scanning the aluminum workpiece at an angle $\theta = 80^\circ$ in the cone-beam X-ray CT, see Figure 6. At this angular position an average deviation from CMM of only $0.1 \mu\text{m}$ is achieved by the CT measurements of diameter of the hole for the aluminum hollow block at the orientation O1. For the PMMA hollow block, the average deviation from CMM for CT measurements of diameter of the hole are less

than 1 μm for the full range of angular variations, with associated standard deviations of about $\pm 5 \mu\text{m}$, in good agreement with the magnitude of the accuracies (and uncertainties) claimed by modern microfocus CT systems.

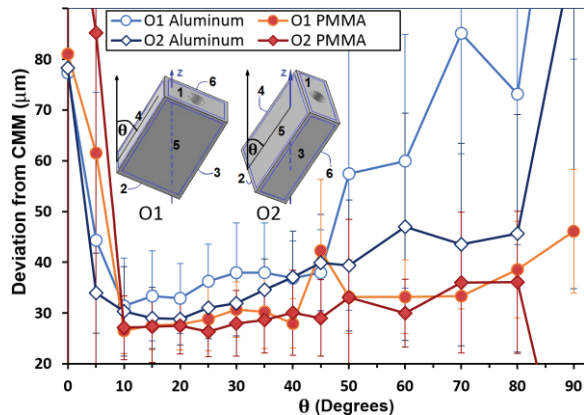


FIGURE 8. Mean error of CT measurements of flatness for the hollow block under variations of axial tilt during scan in cone-beam X-ray CT.

To assess geometrical form, measurements of roundness for the internal hole and flatness for the external faces of the hollow block, under variations of axial tilt for scanning in cone-beam X-ray CT, were performed. The main results are presented in Figure 7 and Figure 8. Inclinations larger than 40 or less than 10 degrees strongly increase the deviations from reference measurements with CMM, with errors up to 80 μm for flatness and more than 100 μm for roundness. But it also seems evident, from Figure 7 and Figure 8, that to get the minimum errors in measurements of form with X-ray CT, the ideal angle tilt for optimal reconstruction of the object's surfaces (for the hollow block workpiece) is around 10 to 20 degrees, for both, the aluminum and PMMA material. This demonstrates that it is possible to find optimal orientations for performing precise dimensional measurements with CT, although in this case, for measurements of form, there still exists a bias in respect to reference CMM of around 30 to 40 μm at the optimal inclinations. However, these deviations are still below the 54 μm voxel size used for scanning the workpiece. For distance measurements (length, width, and diameter), the average CT measurement deviations from CMM can be minimized to less than 1 μm by scanning the workpieces at optimal angles as demonstrated above. The errors in measurements of form with X-ray CT seem to

affect more the measurement outcomes for the aluminum workpiece when compared to the PMMA, most likely because of errors resulting from surface determination in the CT data. These errors might be further minimized by optimizing other parameters of the CT scanner such as the number of projections (the experiments performed in this research collected only 800 projection images per scan), integration time per image, and perhaps the energy dependent settings for the X-rays such as pre-filtration or energy of the beam.

CONCLUSIONS

For the specimens examined in this study, it is concluded that angular positions (axial tilts illustrated in Figure 2) to obtain the small deviations between dimensional measurements between contact CMM and CT scans for the samples studied here lie in range of 10 to 40 degrees, and angular positions above 60 degrees strongly affect the volume reconstruction of the parts creating CT artifacts, hence, lowering the accuracy of CT dimensional measurements. Vertical orientations or angular positions close to 0 (i.e. on the CT beam cone axis and aligned with vertical and horizontal planes) are also not ideal because they cause strong Feldkamp artifacts in the CT image reconstruction process. At the optimal positions/orientations for the sample inside the CT cavity, the accuracies of distance measurements (lengths, widths, and diameters) fell within the claimed uncertainties of around 4 to 8 μm for the modern microfocus CT systems. The average deviations obtained respect to tactile CMMs were less than 1 μm , with associated standard deviations of about $\pm 5 \mu\text{m}$, when scanning the workpieces at convenient angles. This demonstrates that it is possible to find optimal orientations for performing precise dimensional measurements with CT. However, for measurements of form, larger deviations were found, as it is commonly experienced with the CT technique [18-23], mainly due to threshold determination issues associated with edge detection or surface extraction algorithms, but the deviations at the optimal inclinations were still below the voxel size (54 μm) used for scanning the workpiece.

Lastly, the knowledge gained from this study might be useful in the creation of guidelines for the acquisition of CT data for dimensional metrology of components of known shape, and also as a reference for simulation-aided tools in industrial X-ray CT metrology.

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REFERENCES

- [1] Carmignato S. Accuracy of industrial computed tomography measurements: Experimental results from an international comparison. *CIRP Annals – Manufacturing Technology* 61 (491-494), 2012.
- [2] Jiménez R., Ontiveros S., Carmignato S., and Yagüe-Fabra J.A. Fundamental correction strategies for accuracy improvement of dimensional measurements obtained from a conventional micro-CT cone beam machine. *CIRP Journal of Manufacturing Science and Technology*. Vol. 6-2 (143-148), 2013.
- [3] Müller P., Hiller A., and De Chiffre L. A study on evaluation strategies in dimensional X-ray computed tomography by estimation of measurement uncertainties, *IJMQE* 3 (2, 107-115), 2012.
- [4] Weckenmann A., and Kramer P. Assessment of measurement uncertainty caused in the preparation of measurements using computed tomography. XIX IMEKO world congress – Fundamental and applied metrology, September 6-11, 2009, Lisbon, Portugal.
- [5] Feldkamp L. A., Davis J., and Kress J. W., Practical cone-beam algorithm. *J. Opt. Soc. Amer.* 1 A6 (612-619), 1984.
- [6] Tuy H. An inversion formula for cone-beam reconstruction, *SIAM Journal of Applied Mathematics* 43 (546-553) 1983.
- [7] Bartolac S., Clackdoyle F., Noo F., Siewerdsen J. H., Moseley D., and Jaffray D. A. A local shift-variant fourier model and experimental validation of circular cone-beam computed tomography artifacts, *Med. Phys.* 36 2 (500-512), 2009.
- [8] Villarraga-Gómez H., Dimensional Metrology and Optimization of Specimen Orientation in Cone-beam X-ray Computed Tomography, PhD Thesis (to be published), University of North Carolina at Charlotte, USA.
- [9] Balle B., and Zeiler W, Calypso – Visual Metrology. *Innovation Special Metrology* 6 (8-9), The Magazine from Carl Zeiss, 2006.
- [10] Benninger R, METROTOM 800 computer tomograph. *Innovation Special Metrology* 11 (6-7), The Magazine from Carl Zeiss, 2009.
- [11] Lettenbauer H, METROTOM – Measure in a new dimension. *Innovation Special Metrology* 9 ((12-13), Magazine from Carl Zeiss, 2007.
- [12] Carl Zeiss Industrielle Messtechnik GmbH, Bridge-type measuring machines, EN_60_020_1581, Carl Zeiss, Germany.
- [13] Doiron T., A Short History of the Standard Reference Temperature for Industrial Dimensional, *J. Res. NIST* 112 (1-23), 2007.
- [14] Villarraga-Gómez H., Thousand J. D., Morse E. P., Hocken R. J., and Smith S. T. CT measurements and their estimated uncertainty: The significance of temperature and bias determination, *ASPE Mets & Props*, 60, J. Phys.: Conf. Ser. 2015.
- [15] Dewulf W., Tan Y., and Kiekens K. Sense and Non-Sense of Beam Hardening Correction in CT Metrology. *CIRP Annals. Manufacturing Technology*, 61 (1), (495-498), 2012.
- [16] Tan Y., Kiekens K., Welkenhuyzen, F., Angel, J., De Chiffre L., Kruth J., and Dewulf W. Simulation-aided investigation of beam hardening induced errors in CT dimensional metrology. *Measurement Science & Technology*, 25 (6), 16, 2014.
- [17] Tan Y., Kiekens K., Welkenhuyzen F., Kruth J., and Dewulf W. Beam hardening correction and its influence on the measurement accuracy and repeatability for CT dimensional metrology applications. *Conf. on Industrial Computed Tomography*. Wels, Austria, 2012.
- [18] Kiekens K., Tan Y., Kruth J., Voet A., Dewulf W. Parameter dependent thresholding for dimensional X-ray computed tomography. *Proc. of the International Symposium on Digital Industrial Radiology and Computed Tomography*. Berlin, Germany, 2011.
- [19] Tan Y., Kiekens K., Kruth J., Voet A., Dewulf W. Material Dependent Thresholding for Dimensional X-ray Computed Tomography. *International Symposium on Digital Industrial Radiology and Computed Tomography*. Berlin, Germany, 20-22 June 2011.
- [20] Villarraga-Gómez H., Morse E. P., Hocken R. J. and Smith S. T. A study on material influences in dimensional computed tomography. *ASPE Annual Meeting* 59 (684-689), 2014.

- [21] Villarraga-Gómez H., Morse E. P., Hocken R. J. and Smith S. T. Dimensional metrology of internal features with x-ray computed tomography. ASPE Annual Meeting 59 (67-72), 2014.
- [22] Villarraga-Gómez H., Clark D. O., and Smith S. T. Strategies for coordinate metrology on flexible parts: CMM vs CT, ASPE Mets & Props, 60, J. Phys.: Conf. Ser. 2015.
- [23] Léonard F., Brown S. B., Withers P. J., Mummery P. M., and McCarthy M. B. A new method of performance verification for X-ray computed tomography measurements, Meas. Sci. Technol. 25 (1-0), 2014.

IN-LINE, WAFER-SCALE INSPECTION IN NANO-FABRICATION SYSTEMS

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ABSTRACT

This article presents the design of an X-Y precision stage utilizing a double parallelogram flexure mechanism (DPFM) capable of mm-scale displacement in two dimensions. This design is for a single-chip AFM stage and provides precise positioning of the chip for wafer inspection. As long as the geometric dimensions of the flexure are optimized, it is possible to put multiple stages in a narrow space, which enables the simultaneous inspection of multiple areas on a wafer for in-line nanomanufacturing applications.

INTRODUCTION

In the nano-patterning industry, metrology [1] is an important issue because it is needed to ensure the correct alignment and patterning of features. Integrating a sensing system which can instantaneously send back the dimensional information about manufactured products can reduce losses and defect rates. However, it is difficult to perform in-line metrology in nano-fabrication systems because it requires not only real-time inspection but also nanoscale resolution of complex features.

Atomic force microscopy (AFM) has high resolution ability (sub-nm-scale) and is widely utilized in scientific and industrial applications. However, there are two main setbacks to the use of AFM metrology in in-line manufacturing applications [2]. The first is the scanning speed of the AFM tip and the second is the time it takes to place and align samples in the AFM.

The limitations created by the long setup time and the low speed scanning can be solved by using single-chip AFMs [3]. Single-chip AFMs use MEMS-based flexures, sensors, and actuators to scan the AFM tip across a surface in order to achieve high speed measurements with small device dimensions. In the design of the inspection system presented in this paper, a precision XY stage is used to position the single-chip AFM relative to the inspection wafer. The aim of this project is to enable atomic force microscopy to be integrated directly into wafer fabrication lines.

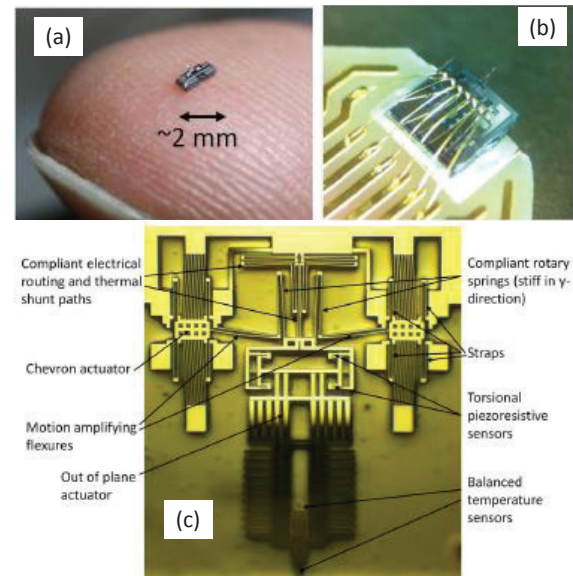


FIGURE 1. (a) A single chip AFM, (b) packaged instrument, and (c) layout of MEMS chip [2]

Flexure bearings are used in this system because of their precision and mechanical simplicity [4, 5]. The flexure bearings are used to position the single-chip AFMs relative to the wafer that is being inspected. The precision positioning stage is designed to be able to achieve mm-scale displacements with micron level precision. Multiple stages with multiple independent AFMs can be incorporated in to the

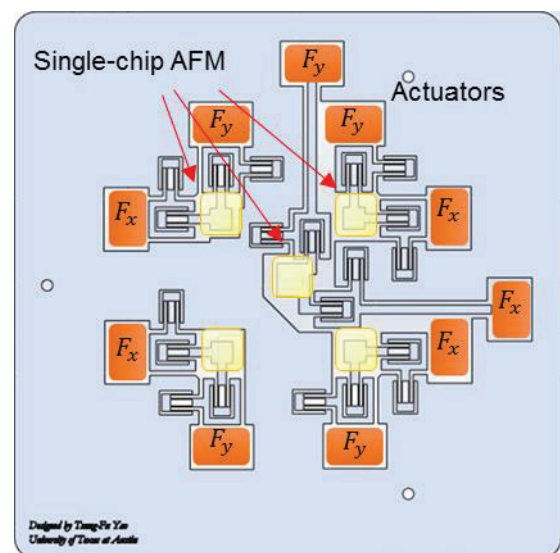


FIGURE 2. Designed XY precision stage for multiple single-chip AFMs inspection

metrology system to increase the inspection speed and area as shown in FIGURE 2. However, in order to be able to incorporate multiple stages into a single inspection system, the size of the positioning flexures must be minimized and the space utilization in the positioning system must be maximized.

PRECISION LINEAR STAGE DESIGN

Flexure mechanisms are commonly used in micro-positioning XY stages due to their superior isolation of motion between the X and Y axis and their great difference between in-plane and out-of-plane stiffness. There are many studies of double parallelogram flexure mechanisms (DPFM), which demonstrate extreme precision with mm-scale displacement range [6-8]. FIGURE 3 shows a simple model of double parallelogram flexure mechanism.

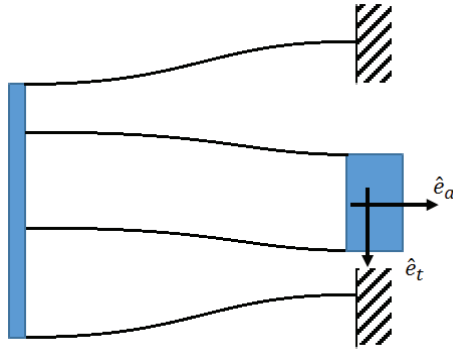


FIGURE 3. Simple model of double parallelogram flexure mechanism

Because of the mm-scale deflection of the flexure beams, stiffness in the axial direction is affected by tangential displacement and tangential stiffness is affected by axial force [6]. That is,

$$K_a \approx \frac{1}{\left(w^2 + \frac{9}{25}x_t^2\right)} \cdot \frac{12EI}{L}$$

and

$$K_t \approx \left[12 - \frac{3}{100} \left(\frac{F_a L^2}{EI}\right)^2\right] \cdot \frac{EI}{L^3}$$

where w , L , and E are the flexure width, beam length, and Young's modulus, respectively and I is the second moment of area of the flexure beam.

FIGURE 4 shows a preliminary design for the XY positioning stage using the double parallelogram flexure mechanism. The image

on the left is the top stage with the AFM chip and image on the right is the wafer sample stage.

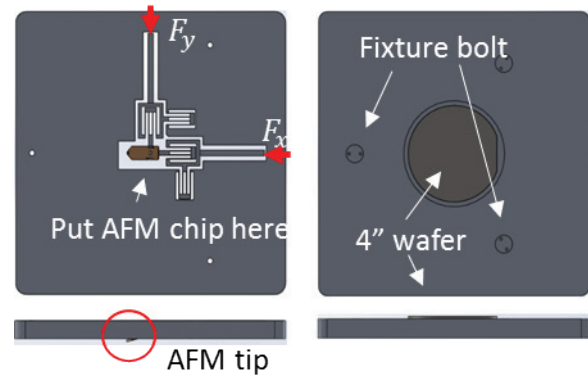


FIGURE 4. XY precision stage with the double parallelogram flexure mechanism

FIGURE 5 is a schematic of the in-line inspection system concept with a flexure stage used position the AFM chip and a sample stage used to hold the wafer for inspection. A kinematic coupling is used to quickly and precisely align the flexure stage to the sample stage.

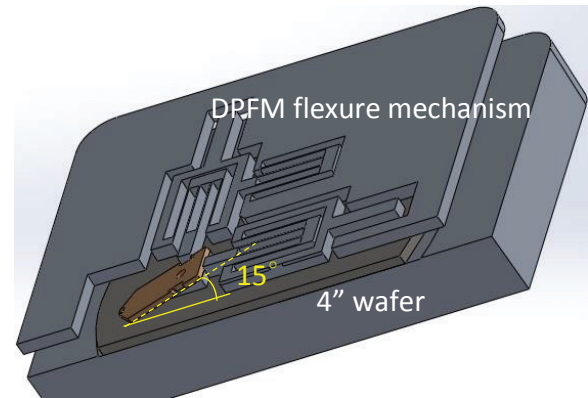


FIGURE 5. Concept for the total wafer inspection stage system (in section view)

The X and Y motions of the flexure are driven by a set of micrometer heads shown as red arrows in FIGURE 4. Each axis of motion can be modeled as a spring system with two parallel sets of springs and two series sets of springs (K_t and K_a). Since stiffness in the axial direction is much greater than stiffness in the tangential direction, total stiffness in the X-direction will be two times the stiffness of each of the DPFMs in the tangential direction. Sensitivity analysis for those multiple variables demonstrates that only variations in the beam length (L) and the flexure width (w) have a significant effect on the

stiffness. Therefore, we can plot the stiffness the stage as a function of the beam length and flexure width, as shown in FIGURE 6.

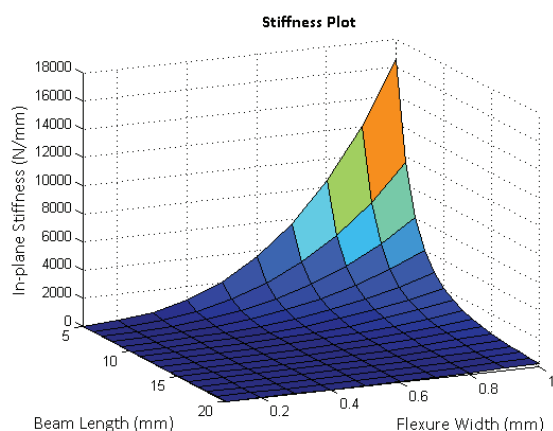


FIGURE 6. Mapping of stiffness varied with beam length (L) and flexure width (w)

In order to provide reasonable stiffness and to minimize the mechanism size, the flexure was designed to be cut from a 15mm-thick block of 7075-T6 aluminum. A length of 20 mm and a width of 0.40 mm was selected for the flexures which resulted in a predicted 19.4 N/mm in-plane stiffness. The first mode natural frequency of the stage was 133 Hz, which is two orders magnitude higher than largest frequencies generated by the laboratory environment. The maximum force input to the mechanism is 50 N

per axis, which results in a 2.58mm displacement of the center stage.

Translation in the Z-direction and rotations about the X and Y axes are accomplished via actuation of three micrometers attached to the AFM stage. The spindle of each micrometer is press-fit to precision truncated balls, shown as FIGURE 7. The balls interface with three vee-blocks and when the micrometers are locked the ball and vee-block coupling kinematically constrains all 6 DOFs of the AFM stage relative to the wafer alignment stage.

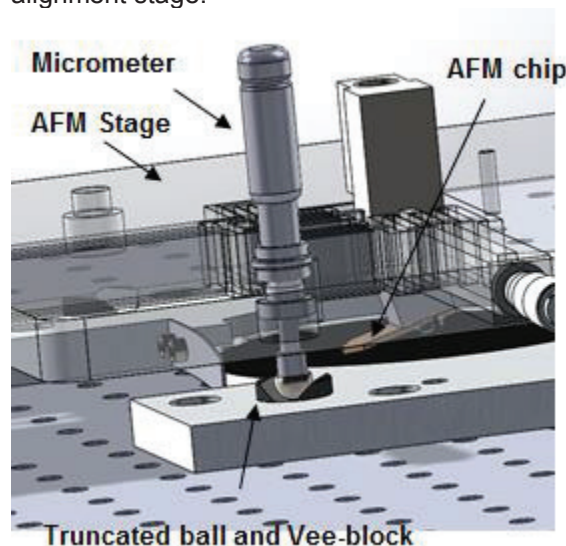


FIGURE 8. A demonstration of how to securely assembly micrometer with truncated balls

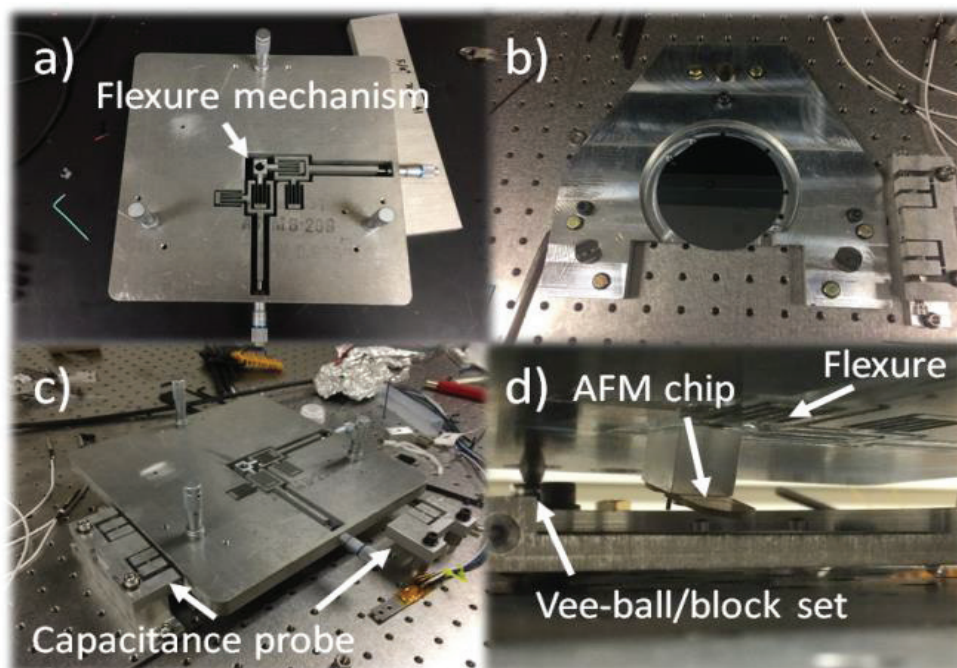


FIGURE 7. Photos show a) flexure-based stage, b) specimen stage, c) stage assembly, and d) AFM chip put underneath of flexure stage

The final flexure-based XY precision stage was machined from a 15-mm thick 7075-T6 aluminum plate using a water jet cutting machine, shown as FIGURE 8.

MEASUREMENT

Asymmetric arrangement of the flexure mechanisms and manufacturing error created in-plane yaw error motion. As a result the motion of the stage differed from the cumulative input from the X and Y actuators. This parasitic motion was measured with two fiber-based optical displacements with 1.0- μm sensitivity and 100-mm working range. The setup is shown in FIGURE 9.

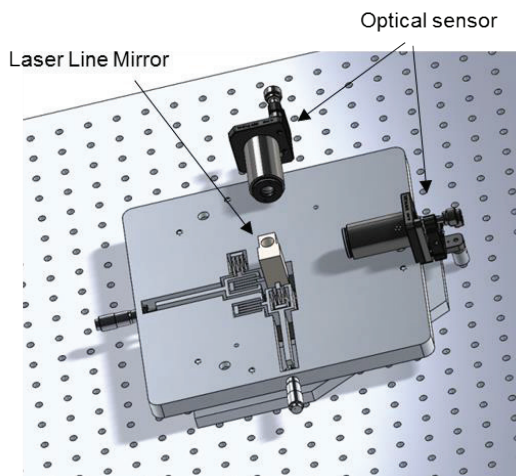


FIGURE 9. Setup for parasitic motion test

Repeatability of the kinematic coupling between the AFM stage and the wafer stage was tested with capacitance probes. Three capacitance probes from LION Precision with 0.14-nm resolution and 2.0-mm working range were mounted on the optical table as the sensors for X, Y, and Z displacements. The experimental setup is shown in FIGURE 10. This setup was used to measure the repeatability in position during the repeated engagement of the kinematic couplings, as the two stages must be separated in order to change samples.

RESULTS

In the flexure motion test, the stage exhibited 1.47- μm deviation in the Y-direction over 100- μm of actuation in the X-Direction and 3.80- μm deviation in the X direction over 100- μm of actuation in the Y-direction. The results indicated that the motion of the flexure-based

stage is not in a set of perfect perpendicular lines, as shown in FIGURE 11. This error was caused by geometric asymmetry and manufacturing errors. However, these errors are repeatable and therefore can be calibrated for in the actuation system.

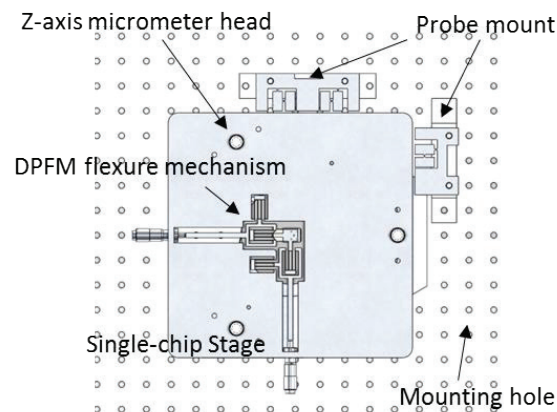


FIGURE 10. Capacitance Probe Setup

Finite Element Analysis (FEA) was used to calculate the parasitic motion without manufacturing error. The results indicated 7.55- μm X-deviation per 100- μm Y-actuator travel and negative 8.31- μm Y-deviation per 100- μm X-actuator travel. The difference between parallelism test and FEA result suggests that manufacturing tolerances contributed to the parasitic motion. Water jet cutting the stage yielded uneven flexure thicknesses which affected the motion of flexure.

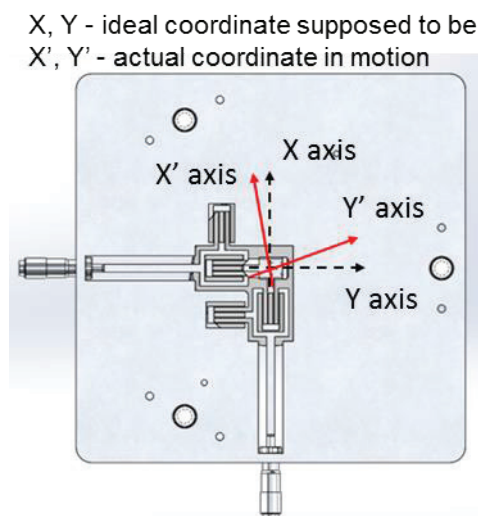


FIGURE 11. Schematic drawing for the single-chip stage parallelism

Results for the repeatability of the kinematic coupling were recorded for translation in X, Y, and Z positions as well as rotated angle about the Z-axis of the stage. FIGURES 12, 13, and 14 show the error distribution for position error over 50 trials in X, Y, and Z respectively. FIGURE 15 shows rotational error about the Z-Axis and Table 1 summarizes the repeatability for each degree of freedom. Repeatability is defined as the standard deviation calculated from the trials. In this study, the flexure-based stage has in-plane repeatability of 350 to 400 nm with in-plane rotation of 0.140 μrad and about 60-nm out-of-plane translational repeatability. Generally, AFM equipment scans in a couple of micron meter range, which is greater than the repeatable error. Therefore, an in-plane error of 400 nm is acceptable to be used in single-chip AFM operation.

Repeatability	X(μm)	Y(μm)	Z(μm)
St. D.	0.390	0.361	0.060

TABLE 1. Repeatability performance in X, Y, Z and rotation of X-Y plane for the flexure-based precision stage

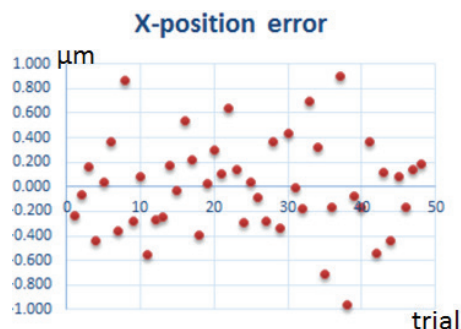


FIGURE 12. X-Axis Translational Error

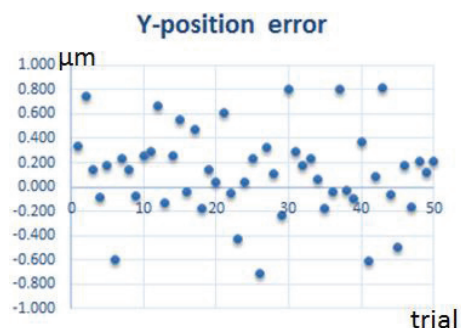


FIGURE 13. Y-Axis Translational Error

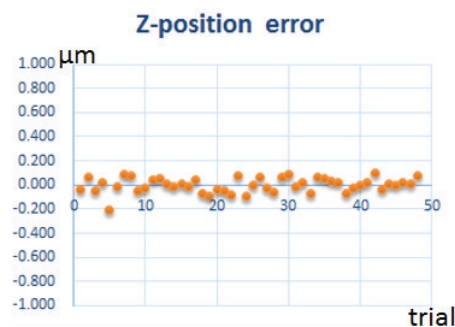


FIGURE 14. Z-Axis Translational Error

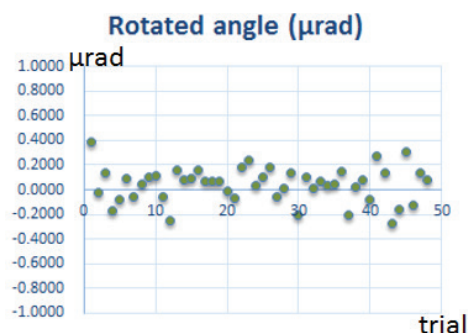


FIGURE 15. Rotational Repeatability Error

CONCLUSION AND FUTURE WORKS

The XY precision stage presented in this article is capable of mm-scale displacement with μm -scale precision and enables the in-line inspection of wafers with single-chip AFMs. Parasitic error when actuating the stage was 1.47 and 3.80 μm deviation in X and Y direction respectively per 100- μm of actuator travel in the orthogonal direction. The kinematic coupling exhibited sub-micron in-plane repeatability of 390 nm and 361 nm, as well as 60-nm precision in out-of plane direction, which are acceptable in operation of AFM scanning.

This flexure mechanism design can be extended to a stage with five independently actuated AFM chips to allow for simultaneous measurements at multiple points on a wafer. In the future, closed-loop feedback control will be implemented to approach the sample in the Z-direction instead of manually turning the micrometer heads. The stage presented in this study, when kinematically coupled to a wafer fixture stage, and automatically actuated in the z-direction will enable in-line metrology of silicon wafers.

REFERENCES

- [1] Morse J, Workshop on Nanofabrication Technologies for Roll-to-Roll Processing, September 27-28, 2011, Boston, MA.
- [2] Sarkar N, Lee G, Mansour RR, Transducers 2013, Barcelona, SPAIN, June 16-20
- [3] Sarkar N and Mansour RR, MEMS 2014, San Francisco, CA, USA, January 26 – 30
- [4] Schitter G, Thurner PJ, and Hansma PK, Mechatronics 2008; 18:282.
- [5] Aphale SS, Bhikkaji B, Moheimani SOR, IEEE Transactions on Nanotechnology 2008; 7(1):79-90.
- [6] Awtar S and Parmar G, J. Mechanisms Robotics 2013; 5(2):021008
- [7] Xu Q, IEEE on Robotics & Automation 2014, Hong Kong, China, May 31 – June 7
- [8] Patil R, Deshmukh S, Reddy YP, and Mate K, ICNTE-2015, Jan 9 – 10

PROPOSAL OF A NOVEL PRESSURE CONTROL METHOD FOR AIR CYLINDERS USED IN HYBRID ELECTRIC-PNEUMATIC ULTRA-PRECISION VERTICAL POSITIONING DEVICE

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INTRODUCTION

The purpose of this research is to propose a novel pressure control method for the balancing air cylinders of a hybrid electric-pneumatic ultra-precision vertical positioning device used in an aspherical lens generator. The positioning device employs a hybrid electric-pneumatic method that combines a pneumatic balancing cylinder(s) as a gravity compensator with a cored electric linear motor in order to achieve both nanometer-level ultra-precise positioning and energy efficiency. Ordinarily, the balancing cylinder pressure is regulated by a diaphragm-type (DH) pressure regulator, which is an extensively used pneumatic element. However, because the dynamic characteristics of DH regulators are not sufficiently high for some applications that require extremely fast and precise responses, pressure fluctuations in the pneumatic balancing cylinder may influence the stage positioning performance, thereby preventing highly precise nanometer-level positioning.

In recent research, in order to clarify the relationship between pressure changes in the balancing cylinder that supports the ultra-precision vertical stage and the positioning performance, a hybrid electric-pneumatic ultra-precision vertical positioning stage was fabricated [1]. In this paper, the realization of a quicker and more precise pressure control method using a new pressure regulator called an HPR (high-precision quick-response pressure regulator), which has been invented by the authors, and a nozzle flapper type servo valve (NF valve) is proposed and discussed.

Firstly, the configuration of the fabricated hybrid electric-pneumatic positioning device is explained. Then, the proposed pressure control method using an HPR and an NF valve is explained. Finally, the proposed method is applied to the pressure control of balancing air cylinders used in a hybrid electric-pneumatic ultra-precision vertical positioning device (a

component of an aspherical lens generator (Fig. 1)).

A HYBRID ELECTRIC-PNEUMATIC POSITIONING DEVICE

As stated in the introduction, the purpose of this research is to realize quick and precise pressure control in the balancing cylinder used in the vertical positioning stage of an aspherical lens generator (Fig. 1). However, since such ultra-precision tools are customarily very complicated, and since numerous factors and conditions are involved, finding a simple method that can be used to clarify these relationships using conventional tools might not be possible.

Therefore, in an attempt to clarify these points, a simple, hybrid electric-pneumatic vertical positioning device that is capable of reproducing the vertical positioning component of an actual aspherical lens generator was fabricated. The structure and appearance of the fabricated device are shown in Figs. 2 and 3. The device is composed of a vertical stage, a coreless-type linear motor (S250D, GHC), two pneumatic balancing cylinders with aerostatic linear bearings, and an absolute-type linear scale (LIC4017, HEIDENHAIN) with a resolution of 1 nm. The specifications of the fabricated device are shown in Table 1.

The special characteristics of this device include its ability to suppress friction, stick-slip, and heat

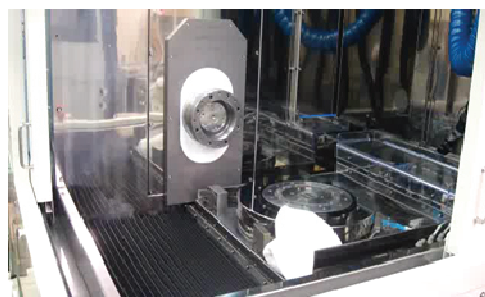


Fig. 1 Stage of aspherical lens generator.

generation due to the adaption of a finely finished balancing cylinder surface with an aerostatic linear bearing and a non-contact (coreless) type linear motor. The system configuration used for operating the device is shown in Fig. 4.

First, the position signal output from the linear encoder is transmitted to the MOVO2 (SVFH3-H3-DSPNMK, Servoland), which functions both as a digital signal processor (DSP) and a driver. The position command is then sent from a computer to the MOVO2. In the MOVO2, the velocity and acceleration values are calculated by the position signal in order to implement PDD2 control, and the control current signal is transmitted to the linear motor. The measured values for the position, pressure, and pressure differential (PD), which will be discussed later, are then saved by another DSP.

THE PROPOSED PRESSURE CONTROL METHOD

Ordinarily, the balancing cylinder pressure is regulated by a DH-type pressure regulator, which is an extensively used pneumatic element. However, because the dynamic characteristics of DH regulators are not sufficiently high for some applications that require extremely fast and precise responses, we previously developed an HPR [2][3] that can manipulate the supply pressure very quickly and precisely. The components and the block diagram of the HPR are shown in Figs. 5 and 6. As indicated in Fig. 6, the main loop is a pressure feedback loop. The differentiated pressure is a feedback parameter in the minor loop. Thus, a PI-D (derivative preceding) control system is used to maintain a constant pressure in the isothermal chamber, irrespective of the upstream or downstream disturbances. The minor loop also reduces the effects of the nonlinear characteristics of the SP valve and the supply pressure fluctuations. In this research, a quicker and more precise pressure control method using the HPR and an NF valve is proposed (Figs. 7, 8).

Table 1 Specifications of the fabricated device

Linear motor	Rated thrust [N]	38
	Rated current [A]	1.3
	Maximum thrust [N]	148
Linear encoder	Encoder type	Absolute
	Resolution [nm]	1
Stage	Weight of movable part [kg]	19.81
	Bearing type	Hydrostatic bearing

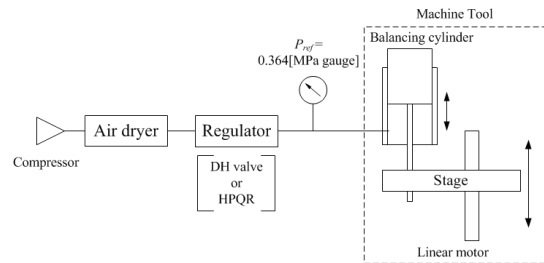


Fig. 2 Schematic of the experimental device (a hybrid electric-pneumatic ultra-precision vertical positioning device).

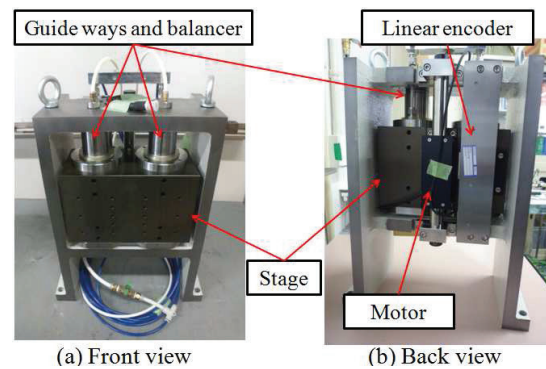


Fig. 3 Fabricated experimental device.

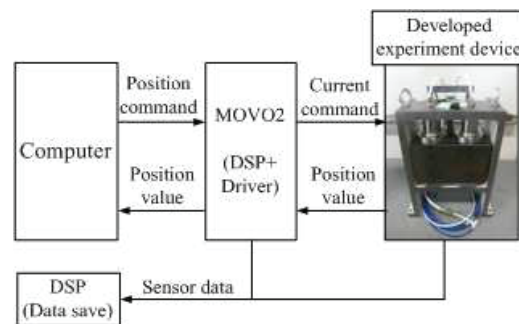


Fig. 4 System configuration.

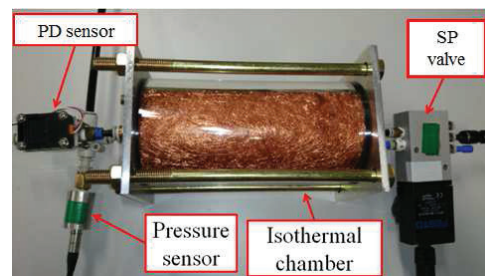


Fig. 5 Components of the HPR.

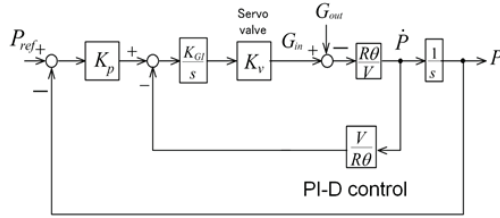


Fig. 6 Block diagram of the HPR.

Experiment using a fixed chamber

Firstly, a feedback control experiment of the initial pressure in the fixed chamber utilizing the proposed method is conducted. The experimental configuration and a diagram are shown in Figs. 9 and 10, respectively. The chamber with a volume of $V=1.5$ L imitates a balancing cylinder in the vertical stage of an aspherical lens generator. In the controller, both PI feedback control and feedforward control are utilized. In the experiments, the PI gains are set as $k_p=11.491 \times 10^{-5}$ and $k_i=0.8772 \times 10^{-5}$. In the experiment, the supply pressure from the compressor to the HPR is set at 500 kPa (gauge).

Two kinds of experiments are conducted with the proposed pressure control method as follows:

- 1) Step response experiment: The pressure value is initially set to $P_{ref}=210$ kPa (gauge), and at $t=30$ s, P_{ref} is changed to 220 kPa (gauge).
- 2) Disturbance experiment: As shown in Fig. 9, a spool-type servo (SP) valve is attached downstream of the fixed chamber. The SP valve provides a disturbance. The spool position is neutral when the control signal is 5.0 V. Initially, the pressure in the chamber is regulated at 220 kPa (gauge), and the control signal of the SP valve is kept at 5.0 V. Then, at $t=40$ s, the control signal of the SP valve is changed to 6.5 V.

The experimental results are shown in Figs. 11 and 12. In Fig. 11, along with the result using the proposed method, the experimental result for the "open loop" is indicated for comparison. "Open loop" means the control current to the NF valve is changed to a step shape (without feedback or feedforward control). As indicated, the time constant required with the open loop is 13 s, whereas with the proposed method, it is only 0.45 s. Next, in Fig. 12, the experimental results for the disturbance experiments are shown. Here, PI+FF is our proposed method, and for comparison, the result with PI control (without

feedforward (FF) control) is indicated. This validates the effectiveness of the proposed feedforward control in terms of avoiding the influence of the disturbance.

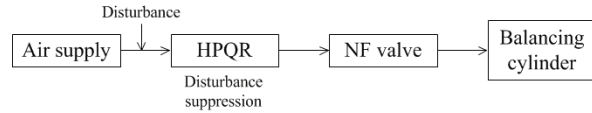


Fig. 7 Schematic of the proposed pressure control system.

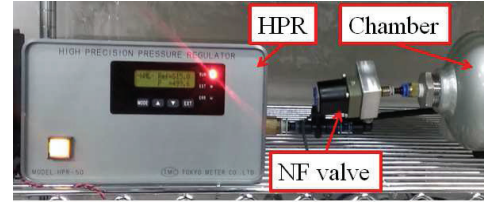


Fig. 8 Proposed pressure control system.

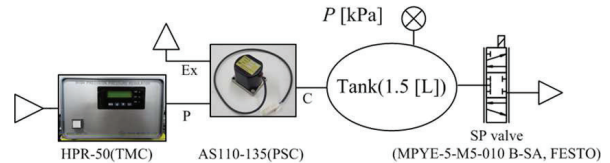


Fig. 9 Experimental configuration of the proposed pressure control method using a fixed chamber.

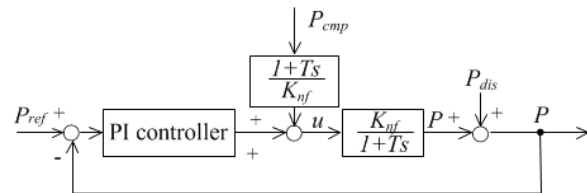


Fig. 10 Diagram of the proposed pressure control method using a fixed chamber.

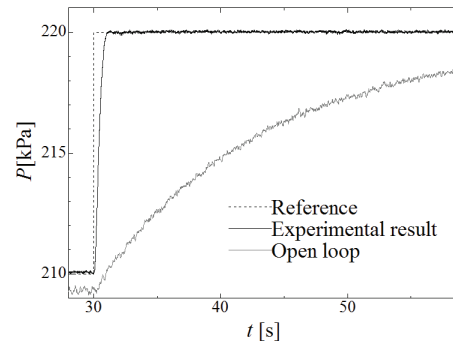


Fig. 11 Experimental results for step responses (using a fixed chamber).

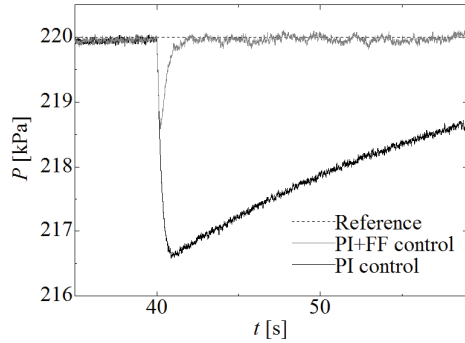


Fig. 12 Experimental results of disturbance experiments (using a fixed chamber).

Experiment using a hybrid electric-pneumatic positioning device

Finally, the proposed method is applied to the pressure control of balancing air cylinders used in a hybrid electric-pneumatic ultra-precision vertical positioning device. A block diagram of the control for a hybrid pneumatic positioning device using the proposed pressure control method is shown in Fig. 13. Here, in order to

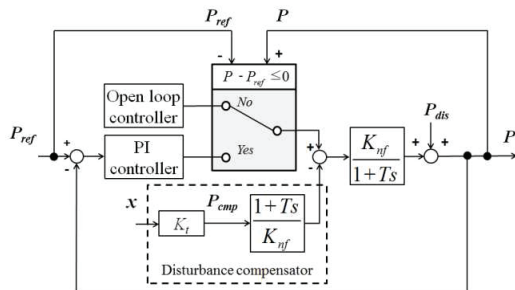


Fig. 13 Block diagram for controlling a hybrid pneumatic positioning device using the proposed pressure control method.

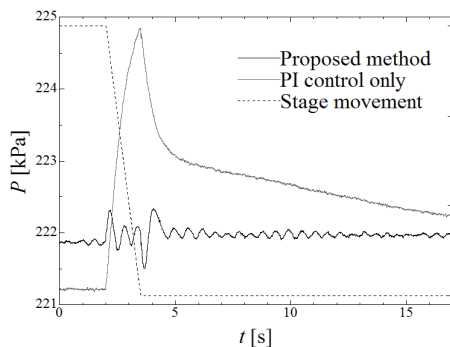


Fig. 14 Experimental results (using the proposed pressure control method).

compensate for hysteresis, an open-loop controller is installed in addition to the proposed pressure control method. In the experiment, the pressure in the balancing cylinder is first set at 220 kPa (gauge), and then, the stage is moved downward at a speed of 12000 mm/min for 30 mm. The experimental results for the pressure in the balancing cylinder are shown in Fig. 14. Here, the result with “PI control only (without open-loop controller or feedforward control) is shown for comparison along with the proposed method. The result indicates the superiority of the proposed method at controlling the pressure in the balancing cylinder of a hybrid electric-pneumatic positioning device. This means that the proposed method can reduce the driving current signal of a linear motor used in the stage and even the lead time for manufacturing.

CONCLUSIONS

In this paper, a novel pressure control method utilizing both the feedback and feedforward methods was proposed. Then, the proposed method was applied to the pressure control of balancing air cylinders used in a hybrid electric-pneumatic ultra-precision vertical positioning device (a component of an aspherical lens generator). Based on the experimental results, the superiority of the proposed control method is demonstrated.

ACKNOWLEDGEMENT

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REFERENCES

- [1] T. Kato, T. Hirakawa, H. Masuda, Pressure control in balancing cylinder for hybrid electric-pneumatic ultra-precision vertical positioning stage, Proceedings of The 6th International Conference on Positioning Technology (ICPT2014), 260-264 (2014).
- [2] T. Kato, K. Kawashima, T. Funaki, K. Tadano, T. Kagawa, A new, high precision, quick response pressure regulator for active control of pneumatic vibration isolation tables, Precision Engineering, Vol. 34, No. 1, 43-48 (2010).
- [3] T. Kato, G. Higashijima, T. Yazawa, T. Otsubo, Y. Nozaki, K. Tanaka, Rotation control and disturbance force compensation of air turbine spindle for precision machining using high-precision, quick-response pneumatic pressure regulator, Proceedings of 2012 ASPE Annual Meeting, Vol. 54, 169-172 (2012).

MODELING AND OPTIMIZATION OF MEMS THERMAL ACTUATOR

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ABSTRACT

This paper presents a structural optimization method of Micro-electro-mechanical systems (MEMS) thermal actuator for a maximum displacement output. MEMS thermal actuator has the capability of producing large actuation force with small footprint size. In this paper, we derive an analytical stiffness model by applying the Screw Theory. Based on the loading condition, we present the actuation models of the thermal actuator. In this paper, we propose a method to simplify the actuation model and to obtain the optimal angle setting for the improved performance of the larger output displacement and improved sensitivity.

INTRODUCTION

An actuator is a type of device to move or control a mechanism or system by converting various types of energy into motion. Based on the actuation output, actuators are categorized by producing force or moment. Based on the types of motion, actuators are categorized as translational and rotational displacement actuators. For example, a rotational electrical motor converts electrical power to rotational displacement with a moment. When a lead screw is combined with the motor, it is converted to a linear actuator [1] by transfer the rotational displacement to a translational displacement [2, 3]. Pneumatic cylinder and piezoelectric actuator are usually used as linear actuators.

In order to achieve a high precision motion control, a lot of current meso-scale motion stages adopt the piezoelectric actuators. Piezoelectric actuators are featured for a small size, high precision, but they have a small stroke and hysteresis phenomenon. Furthermore, piezoelectric actuators are limited in the usage of meso-scale device. In the design of microelectromechanical systems (MEMS), the mostly used actuators are capacitive actuators and thermal actuators. Capacitive

actuators, also called comb drives, can produce relatively large stroke with quick response. The control algorithm of comb drive is relatively easy to build. However, the produced actuation force is relatively low. In order to produce the same force as a thermal actuator, comb drive requires a much larger foot print. Thermal actuators are the other devices widely used in MEMS mechanism. They have the advantages of producing large force with small size, but they are not easy to layout as comb drive. The design of a thermal actuator is also more complicated due to the thermal distribution and the isolation. According to the deformation of the thermal actuator beams, the thermal actuators are categorized by the deformation of radial bending and axial elongation.

A lot of research has been done in design, analysis and application of the MEMS thermal actuators. Yan et al. [4] designed, modeled and fabricated a MEMS bidirectional vertical thermal actuator. Kurita et al. [5] simulated and fabricated MEMS thermal actuator for active flying-height control slider. Kapels et al. [6] applied a thermal actuator in the analysis of fracture strength and fatigue of polysilicon. Sinclair [7] designed a MEMS thermal actuator which is featured for a high force output with a low area footprint. Hubbard and Howell [8] designed a stage using four thermal actuators. Shi et al. [9] did an optimization of a hexapod nanopositioner which is actuated by six thermal actuators. Yong et al. [10] designed and analyzed the thermal actuator according to buckling status. However, there is less work done in the modeling and optimization of the thermal actuator.

In this paper, we propose an optimization algorithm of MEMS axial thermal actuator to reach a maximum displacement. We firstly present the fabrication of thermal actuator. The actuation force is simplified based on the geometric outline. An analytical model is then derived by applying

the Screw Theory to obtain the stiffness of the actuator. A general symbolic actuation model of motion is presented for both the free end and the loading condition. Based on the actuation model, the optimization algorithm is proposed for the optimal angle to reach the maximum displacement.

DESIGN AND FABRICATION

As shown in Figure 1, the thermal axial actuator is composed of three parts: side beams, electrical pads, and center actuation bar. Dozens of the side beams are located on the two sides of the center thermal actuation bar. Two electrical pads are used to connect outside electrical circuit. When the power supply produces the voltage on the pads, the current will go through the side beams. Because of the electrical resistance of the side beams, a mount of heat is produced through the physical phenomena, joule heating. The created heat transfers through the beams and the distributed heat in the beam results in the thermal structural deformation of each side beam. Each beam is elongated due to the thermal deformation. The side beams are connected in parallel to the center actuation bar to create a symmetrical layout. The deformation of each beam produces an actuation force along the center bar.



FIGURE 1. MEMS thermal actuator

In the vertical direction, a thermal actuator usually has four layers: top metal layer, silicon device layer, oxide layer and bottom silicon handle layer. The thickness of the device layer determines the out-of-plane stiffness of the structure. Thus, a thicker device layer increase the survivability of the fabrication and the allowed out-of-plane loading on the device. In this paper, we use a silicon - insulator (SOI) wafer with a $30\ \mu\text{m}$ device layer, $400\ \mu\text{m}$ bottom layer and $2\ \mu\text{m}$ insulator layer. As shown in Figure 2, the buried oxide layer is an electrical isolating layer, which blocks the electrical transmission. The trench shown in Figure 2 is built in the device layer to isolate the current transmitting in the top layer. This ensures that the temperature is only generated at the designed position. By the adoption of the trench, we maximize the deformation of the thermal actuator and minimize the thermal expansion of the loading structure attached to the thermal actuator. The han-

dle layer with $400\ \mu\text{m}$ in depth is a fixed frame to suspend the device layer. The device layer is deformable independently from handle layer. We apply deep reactive ion etching (DRIE) to fabricate the thermal actuator from both sides of the wafer in this paper.

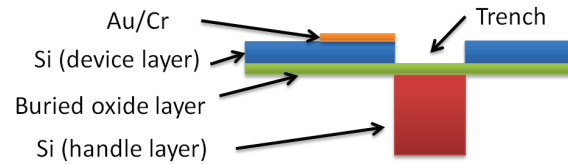


FIGURE 2. Fabrication of MEMS thermal actuator

ACTUATION FORCE

The schematic drawing of the thermal actuator is shown in Figure 3.

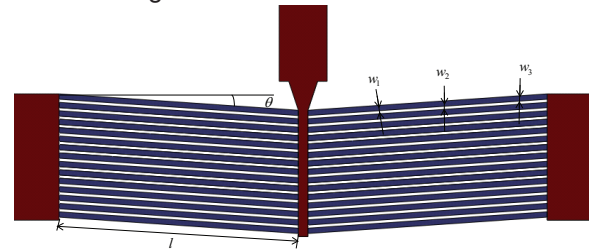


FIGURE 3. Schematic drawing of the thermal actuator

The thermal deformation of each side beam is related to the factors: geometric dimension, the material properties, and thermal distribution. The thermal deformation produces the force of each side beam. Furthermore, the forces created by each side beam are exerted on the center actuation bar. The actuation force on the actuation bar is calculated as

$$F^A = 2\alpha n \bar{T} E w_1 h \sin \theta, \quad (1)$$

where α , n , $\bar{T}$, E , w_1 , h , and θ are physical and material geometric parameters of the thermal actuators defined in Table 1 [11].

As shown in Eq. 1, the thermal actuation force is a monotonic linear function of the thermal expansion coefficient, temperature, number of the beams, cross section, and Young's modulus. When we define these parameters as the constant values, the actuation force is simplified to a function of $\sin \theta$.

$$F^A = \beta \sin \theta, \quad (2)$$

TABLE 1. Design parameters of the thermal actuator.

Symbol	Design Parameter	Values
h	Thickness of the device layer	$30\ \mu m$
θ	Angle of thermal actuator beam	$0.068\ rad$
l	Length of thermal actuator beam	$1000\ \mu m$
w_1	Width of thermal actuator beam	$22.3\ \mu m$
w_2	Distance between two thermal actuator beams	$12.648\ \mu m$
w_3	Width of thermal actuator beam along actuator bar	$22.35\ \mu m$
n	Number of thermal actuator beams on each side	15
E	Young's module	130 GPa
ν	Poissons ratio	0.28
α	Coefficient of thermal expansion	$3 \times 10^{-9}T + 3 \times 10^{-6} (^{\circ}C^{-1})\ \dagger$
$\bar{T}$	Actuator average temperature	$< 550^{\circ}C$

$\dagger$: data source [12]

where β is the coefficient of the simplified parameters.

STIFFNESS DERIVATION

After obtaining the equation of actuation force, we need to derive the stiffness model to further derive the actuation model of the thermal actuator. In this paper, we apply the Screw Theory in the derivation of the stiffness matrix [13]. In the Screw Theory, a flexure mechanism is modeled to the subsequent serial and parallel mechanisms.

According to the Screw Theory, we model the topology of the thermal actuator in Figure 4. In the model, the thermal actuator consists two types of segments: red rigid segment and blue flexible segment. The actuation bar is simplified as a rigid segment. The side beams are modeled as a combination of the flexural and the rigid segments. The two ends of the side beam are simplified as the red rigid triangles. The center segment of the side beam is modeled as a long flexible beam with a rectangular cross section. The compliance matrix of one beam with a rectangular cross section is denoted as $[C^b]$. The analytical model of the beam compliance matrix is shown in the Appendix. According to the topology, the side beams are in parallel connected to the center actuation bar. By applying the equation of parallel connection [14], we can derive the compliance matrix of the thermal actuator, $[C^A]$. The coordinate system is show in Figure 4.

The derived compliance matrix of the thermal actuator, $[C^A]$ includes the main compliance element, and the other elements. The main compliance element denotes the relationship of the actuation force and the corresponding translational displacement. $[C^A]$ is an analytical model of the design parameters, $L, h, w_1, w_2, w_3, n, E, \nu, \theta$.

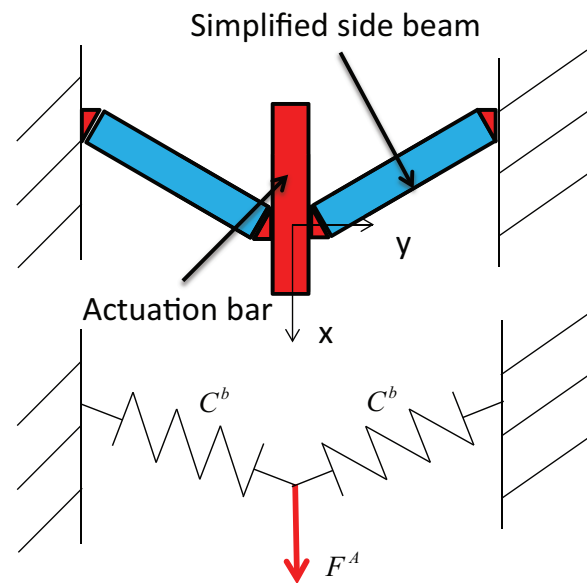


FIGURE 4. Stiffness model of the thermal actuator.

Here, we define θ as the independent variable, the compliance matrix is simplified as a matrix function of θ .

DERIVATION OF THE ACTUATION MODEL

In this section, we derive the analytical model of the motion of the thermal actuator, which includes the derived stiffness and the actuation force. The element of the compliance matrix at the forth row and the first column is denoted as $[C^A]_{41}$. It represents the displacement in the x direction with respect to the force applied in the x direction. The actuation force F^A on the center bar is the summation of the thermal actuation forces of each side beam. When the thermal actuator is without

loading, the actuation model can be written as

$$\delta^A = [C^A]_{41} F^A \quad (3)$$

By combining Eq.1 into Eq.3, we can derive the equation as

$$\delta^A = [C^A]_{41} \beta \sin \theta \quad (4)$$

Since β is a linear equation of the design parameter variables, we simplify the actuation equation in the derivation of the maximum displacement as

$$\delta^{A0} = [C^A]_{41} \sin \theta \quad (5)$$

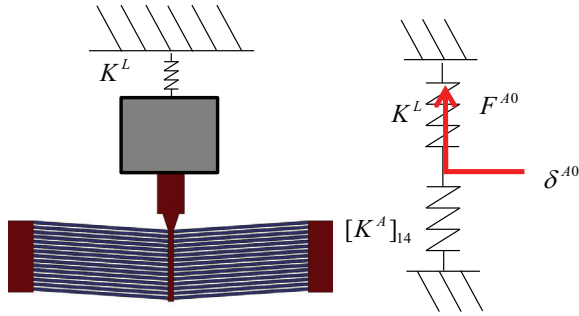


FIGURE 5. Actuation model of the thermal actuator with loading

As shown in Figure 5, the thermal actuator is attached to a target object. The target object is modeled as a rigid body with the stiffness K^L . The actuation model with the loading is derived as

$$\delta^{A0} = \frac{\sin(\theta)}{[K^A]_{41} + K^L} \quad (6)$$

OPTIMIZATION METHOD

In this section, we present the optimization method of the thermal actuator to obtain a large displacement output. Here, we define the θ as a variable. We use an example to explain the optimization. After applying the default parameters shown in Table 1, we can obtain the main element of the stiffness matrix as

$$[K^A]_{14} = 1.3 - 1.3\cos(\theta)^2 + 1.3\sin(\theta)^2 \quad (7)$$

We can find that Eq. 7 is a nonlinear function of the variable, θ . When we plot Eq. 5, we can find the change of the output displacements related to the change of θ in Figure 6. Moreover, we can find that the maximum displacement is $8.59 \mu\text{m}$ when the angle θ equals to 0.025 radians . This θ is the optimal value corresponding to the maximum displacement of the thermal actuator without loading.

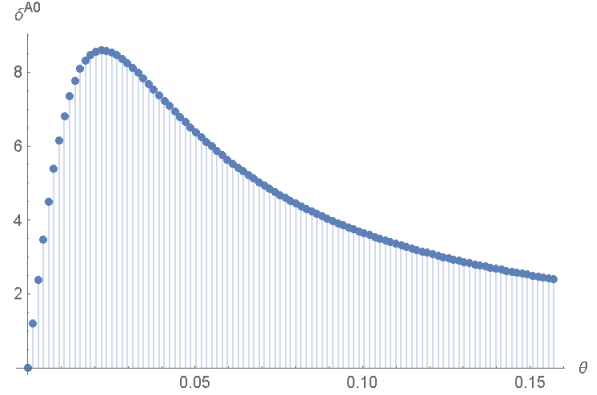


FIGURE 6. Motion of the actuation bar versus the design angle.

Furthermore, we can also plot Eq. 6 after substituting the design parameters as the values in Table 1. Here, θ and K^L are the independent variables and we can plot a three dimension mapping shown in Figure 7. Here, θ ranges from 0 to $\pi/20$. K^L ranges from 0 to $1/100 \text{ N}/\mu\text{m}$. The number of grids is 100. We normalize the two independent variables according to the limit up of the range respectively. The mapping is shown in Figure 7. Given the loading stiffness of the target object, we can find the corresponding 2D drawing of the independent variable θ and the dependent variable δ^{A0} , which is similar with Figure 5. We can obtain the maximum δ^{A0} and the corresponding optimal angle θ .

$$\theta = \pi/20, K^L = 1/100 \quad (8)$$

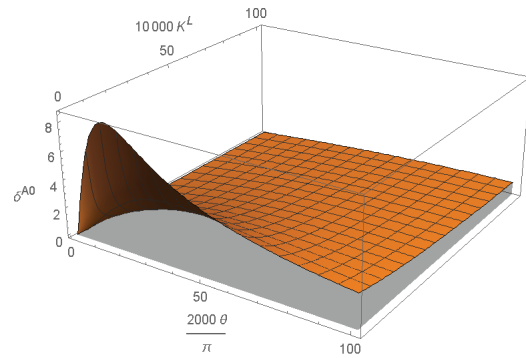


FIGURE 7. Motion of the actuated target object versus the design angle and the loading stiffness.

CONCLUSION

In this paper, we present an optimization algorithm of the thermal actuator for the improved mechanical performance. We firstly present the design and fabrication of a MEMS thermal actuator. Based on the geometric layout, we presented

the analytical model of the actuation force. By means of the Screw Theory, we derived an analytical model of the stiffness matrix of the actuator, of which the side beams are connected to the center actuation bar in parallel. We further derive the actuation model based on the derived actuation force, the stiffness of the thermal actuator, and the stiffness of loading object. The optimal tilt angle of the beams are obtained based on the different loading condition for a maximum output displacement. When the thermal actuator has a larger output displacement corresponding to the same input voltage, the sensitivity of the thermal actuator is improved.

APPENDIX

The compliance matrix of a flexure with rectangular cross section [15] is written as

$$[C^b] = \frac{l}{EI_z} \begin{bmatrix} 0 & 0 & 0 & \frac{1}{\chi\beta} & 0 & 0 \\ 0 & 0 & -\frac{l\kappa}{2} & 0 & \kappa & 0 \\ 0 & \frac{l}{2} & 0 & 0 & 0 & 1 \\ \frac{l^2\eta}{12} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{l^2}{3} & 0 & 0 & 0 & \frac{l}{2} \\ 0 & 0 & \frac{l^2\kappa}{3} & 0 & -\frac{l\kappa}{2} & 0 \end{bmatrix}, \quad (9)$$

where

$$\kappa = \frac{I_z}{I_y} = \frac{t^2}{w^2}, \quad \beta = \frac{J}{I_z}, \quad \eta = \frac{t^2}{l^2}, \quad \chi = \frac{G}{E} = \frac{1}{2(1+\nu)}, \quad (10)$$

are non-dimensional constants determined by geometries and material properties. I_y , I_z are the area moments of inertia about y and z axis. J is the polar moment inertia about x axis. E and G are Young's modulus and shear modulus, respectively, and ν is the poisson's ratio. β is the ratio of torsion constant over moment of inertia. For a rectangular cross section, β is defined by

$$\beta = 12 \left(\frac{1}{3} - 0.21 \frac{t}{w} \left(1 - \frac{1}{12} \left(\frac{t}{w} \right)^4 \right) \right). \quad (11)$$

REFERENCES

- [1] Shi H, Su HJ, Dagalakis N, Kramar JA. Kinematic modeling and calibration of a flexure based hexapod nanopositioner. *Precision Engineering*. 2013;37(1):117 – 128.
- [2] Shi H, Su HJ. Workspace of a Flexure Hexapod Nanopositioner. In: *Proceedings of ASME IDETC/CIE*. Chicago, Illinois, USA; 2012, August 12-15. p. 341–349.
- [3] Shi H, Su HJ. An Analytical Model for Calculating the Workspace of a Flexure Hexapod Nanopositioner. *ASME Journal of Mechanisms and Robotics*. 2013;5(4):041009.
- [4] Yan D, Khajepour A, Mansour R. Design and modeling of a MEMS bidirectional vertical thermal actuator. *Journal of Micromechanics and Microengineering*. 2004;14(7):841.
- [5] Kurita M, Shiramatsu T, Miyake K, Kato A, Soga M, Tanaka H, et al. Active flying-height control slider using MEMS thermal actuator. *Microsystem Technologies*. 2006;12(4):369–375.
- [6] Kapels H, Aigner R, Binder J. Fracture strength and fatigue of polysilicon determined by a novel thermal actuator [MEMS]. *Electron Devices, IEEE Transactions on*. 2000 Jul;47(7):1522–1528.
- [7] Sinclair MJ. A high force low area MEMS thermal actuator. In: *Thermal and Thermo-mechanical Phenomena in Electronic Systems, 2000. ITherm 2000. The Seventh International Conference on*. vol. 1; 2000. p. 132.
- [8] Hubbard NB, Howell LL. Design and characterization of a dual-stage, thermally actuated nanopositioner. *Journal of Micromechanics and Microengineering*. 2005 Aug;15(8):1482.
- [9] Shi H, Duan X, Su HJ. Optimization of the workspace of a MEMS hexapod nanopositioner using an adaptive genetic algorithm. In: *Robotics and Automation (ICRA), 2014 IEEE International Conference on*. Hong Kong, China; 2014, May 31-June 7. p. 4043–4048.
- [10] Yang SH, Kim YS, Yoo JM, Dagalakis NG. Microelectromechanical systems based Stewart platform with sub-nano resolution. *Applied Physics Letters*. 2012;101(6):061909.
- [11] Shi H, Su HJ, Dagalakis N. A stiffness model for control and analysis of a MEMS hexapod nanopositioner. *Mechanism and Machine Theory*. 2014;80:246 – 264.
- [12] Kim YS, Yoo JM, Yang SH, Choi YM, Dagalakis NG, Gupta SK. Design, fabrication and testing of a serial kinematic MEMS

XY stage for multifinger manipulation. *Journal of Micromechanics and Microengineering*. 2012;22(8):085029.

- [13] Su HJ, Shi H, Yu J. Analytical Compliance Analysis and Synthesis of Flexure Mechanisms. In: *Proceedings of ASME IDETC/CIE*. Washington, DC, USA; 2011, August 28-31. p. 169–180.
- [14] Shi H. *Modeling and Analysis of Compliant Mechanisms for Designing Nanopositioners*. The Ohio State University. Columbus, OH, USA; 2013.
- [15] Su HJ, Shi H, Yu J. A Symbolic Formulation for Analytical Compliance Analysis and Synthesis of Flexure Mechanisms. *ASME Journal of Mechanical Design*. 2012;134(5):051009.

DESIGN OF A PARALLEL MEMS POSITIONING STAGE WITH DECOUPLED MOTION

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ABSTRACT

This paper presents a novel design of a parallel micro-electro-mechanical systems (MEMS) positioning stage with 2 degrees of freedom (DOF). It is challenging in the the design of the parallel mechanisms with decoupled motions in MEMS version due to the limits of the fabrication, the material properties, and the actuators. This XY positioning stage design is featured for high precision, compact footprint, decoupled motion, and large out-of-plane stiffness. By means of MEMS fabrication, the footprint of the XY positioning stage is only $4.5\text{ mm} \times 4.5\text{ mm}$. However, the positioning stage is capable of reaching a stroke about $10\text{ }\mu\text{m} \times 10\text{ }\mu\text{m}$ with decoupled motions.

INTRODUCTION

A positioning stage is a high precision positioning device used in motion control. Positioning stages are actuated to manipulate an object to a target position. The criteria to evaluate a positioning stage are the size of the device, the range of motion, accuracy, precision, the maximum allowable load, and the response time. By means of Micro-electro-mechanical systems (MEMS) technologies and flexure mechanism, we can build a MEMS mechanism with a significant small footprint and high precision.

There has been a lot of work done in design, fabrication of MEMS positioning stage. Fowler et al. [1] used electrostatic comb drives to actuate a 2DOF MEMS stage for the application in Atomic force microscopes (AFMs). Shi et al. [2, 3] studied a control model of a serial 2DOF positioning stage. Compared to a serial design, parallel mechanism has a smaller size. However, it is difficult to design a parallel mechanism which could has decoupled motion of 2DOF, small size and high out-of-plane stiffness. In this research, we design a parallel kinematic MEMS positioner,

shown in Figure 1, which is composed of two main parts: thermal actuator and parallel guide mechanism.

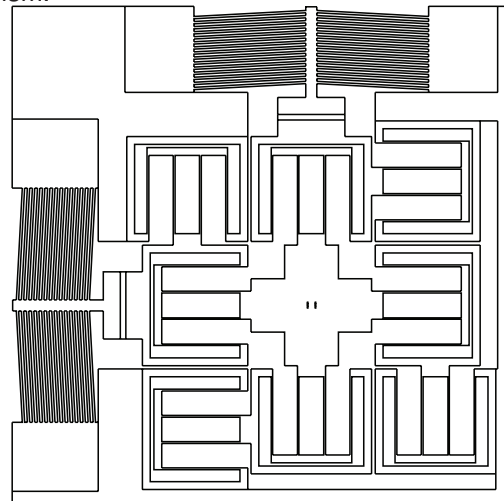


FIGURE 1. Top view of a parallel MEMS positioning stage.

THERMAL ACTUATOR

Thermal actuator is widely used in MEMS mechanism. As shown in Figure 2, there are two electrical pads on both sides to connect to the outside electrical circuit. When the power supply produces the voltage on the pads, the current will go through the beams. Because of the resistance, a mount of heat is created by the physical phenomena, joule heating. The heat will then transfer through the beam by conduction. The distributed heat in the beam will result in the thermal structural deformation of each beam. Finally, the beams in parallel produce an actuating force on the center bar, which is calculated as

$$F_{act} = 2\alpha n \Delta T_{average} E W_b^t T \sin \theta^t, \quad (1)$$

where F_{act} is the output force, α , n , $\Delta T_{average}$, E , W_b^t and θ^t are design parameters, material prop-

erties.

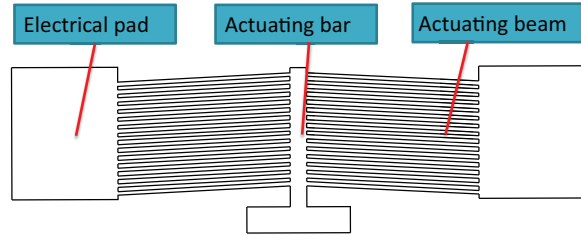


FIGURE 2. Thermal actuator.

GUIDE DESIGN

In the design of the parallel MEMS positioning stage, the guide mechanism is the main component that is used to minimize the footprint, and decouple the translational motions in the x and y directions, and increase the out-of-plane stiffness. Compared to the serial mechanism, kinematic cross error is usually introduced to the parallel mechanism. Therefore, a well designed parallel mechanism is able to reduce the cross error to a very small amount for the multi-axis motion.

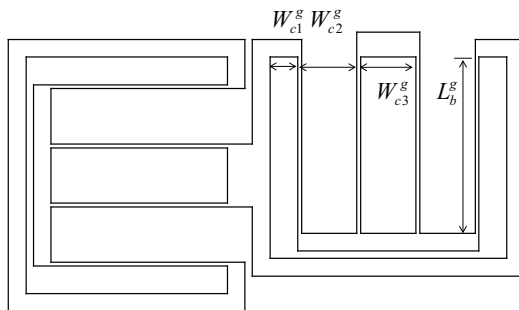


FIGURE 3. Two serial connected guide.

As shown in Figure 3, two beams are firstly connected in parallel to constrain 5DOF, and only one translational displacement output exists. The serial connection of two sets of the two parallel beams makes a basic parallel guide, which is able to produce a larger translational displacement. A basis parallel has 1DOF. Furthermore, a serial connection of two basic guides makes a 2DOF mechanism. This 2DOF mechanism can do decoupled translational motions in the x and the y directions. Finally, four sets of the serial mechanism are located symmetrically in order to reduce the cross error. Another consideration in this design is the out-of-plane stiffness which determines the survivability of the positioning stage during the fabrication process. The four sets of the serial mechanisms allow eight beams to be connected to the frame at the same time, as shown in Figure 1. This shows a higher out-of-plane stiffness of this parallel mechanism than that of a

serial mechanism.

FABRICATION

The proposed MEMS XY stage is fabricated by deep reactive ion etching (DRIE) from both sides of a silicon-on-insulator (SOI) wafer. As shown in Figure 4(b), the wafer has four layers. The buried oxide layer, $2\ \mu\text{m}$, is also an electrical isolating layer. The trench is used to isolating the current from the displacement amplifier so there is no thermal deformation except the actuating beams. The handle layer with $400\ \mu\text{m}$ in depth is used as a fixed frame, which is the red grey bottom in Figure 4(a). The top red part is the device layer in Figure 4(a), which has can be $30\ \mu\text{m}$ to $100\ \mu\text{m}$. Thus, only the top device layer is moveable.

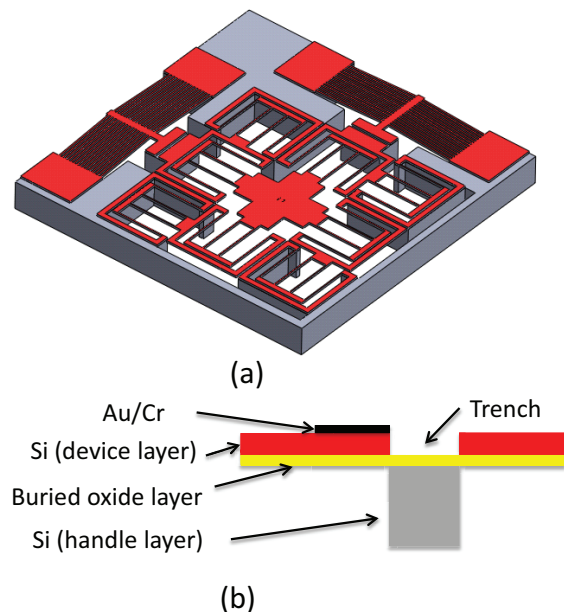


FIGURE 4. Fabrication of the positioning stage with SOI wafer.

REFERENCES

- [1] Fowler AG, Laskovski AN, Hammond AC, Moheimani SOR. A 2-DOF Electrostatically Actuated MEMS Nanopositioner for On-Chip AFM. *Microelectromechanical Systems*, Journal of. 2012 Aug;21(4):771–773.
- [2] Su HJ, Shi H, Yu J. A Symbolic Formulation for Analytical Compliance Analysis and Synthesis of Flexure Mechanisms. *ASME Journal of Mechanical Design*. 2012;134(5):051009.
- [3] Shi H, Su HJ, Dagalakakis N. A stiffness model for control and analysis of a MEMS hexapod nanopositioner. *Mechanism and Machine Theory*. 2014;80:246 – 264.

NANOPOSITIONING STAGES DESIGN FOR FOUR-CRYSTAL HARD X-RAY BEAM SPLIT-AND-DELAY LINE WITH COHERENCE PRESERVATION

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INTRODUCTION

The split-and-delay line is one of the promising technical approaches to study complex nanoscale dynamics in condensed matter in a broad dynamic range from femtosecond to nanosecond regimes by x-ray photon correlation spectroscopy (XPCS) using coherent x-rays from x-ray free-electron lasers (XFELs) [1] and diffraction-limited synchrotron radiation sources, such as the proposed Advanced Photon Source (APS) upgrade project [2]. With an x-ray split-and-delay line instrument, each x-ray pulse is split into reference and delayed pulses of equal intensity, arriving at the sample separated in time. An area detector collects the scattering signals from the reference and delayed pulses during the same exposure.

In 2013, Yuri P. Stetsko and et al. introduced a new split-delay x-ray scheme with a minimal number of x-ray diffracting crystals [3]. Compared with the original split-delay x-ray schemes [4-6], the minimal amount of optical elements simplified the instrument alignment and operations, and also improved the split-delay line efficiency. However, besides designing and fabricating of high quality diamond crystals for beam splitting with coherence preservation, there are many mechanical design challenges in transferring the optical scheme to a practical instrument. First of all, a set of four holders with multidimensional nanopositioning stages for diamond crystals handling and aligning must be designed to meet the challenging design requirements. To minimize the design effort with affordable manufacturing cost, the multidimensional nanopositioning stages were designed with customized modular flexure mechanisms.

As shown in figure 1, the four sets of multidimensional nanopositioning stages (1-4) for the hard x-ray split-and-delay line instrument

are mounted on a vertical mounting base (5) sealed with a plexiglass cover (6) to maintain a helium gas environment with one atmospheric pressure. Each set of the stage system provides two angular (pitch and roll) and two linear (perpendicular to the x-ray beam) adjustment to one of the diamond crystal holders (7-10). The diamond crystal holder (8) has an extra linear stage (11) along its x-ray beam direction to adjust the x-ray delay time. Figure 2 shows a schematic diagram of the x-ray optical path for the prototype of the APS Z8-63 hard x-ray beam split-and-delay line instrument.

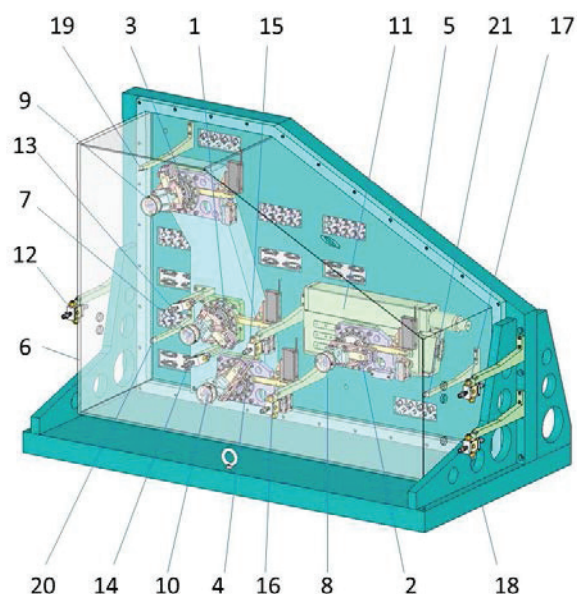


Figure 1. A 3-D model of a prototype of the APS Z8-63 hard x-ray beam split-and-delay line instrument: (1-4) customized multidimensional nanopositioning stages; (5) vertical mounting base; (6) plexiglass cover with vacuum seal; (7-10) diamond crystal holder with individual temperature control; (11) precision linear stage for x-ray delay time control; (12-18) ion chamber; (19-21) YAG crystal screen for x-ray diagnostic.

Mechanical design of the x-ray diamond crystal holder and customized modular flexure stages are discussed in this paper.

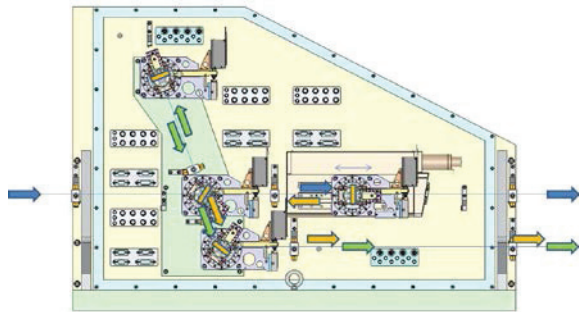


Figure 2. A schematic diagram of the x-ray optical path for the prototype of the APS Z8-63 hard x-ray beam split-and-delay line instrument.

THE DIAMOND CRYSTAL HOLDER

As shown in Figure 3, ultra-thin diamond crystals, which can be up to 30 μm thin, are mounted on CVD diamond base with a special “all-diamond” clamping structure (not shown in this figure) [7]. The CVD diamond base is clamped on an oxygen-free copper (OFHC) revised-T-shaped holder and hosted by a round PEEK base and covered with a plastic enclosure for thermal shielding. There are electric feedthroughs on the PEEK base for a heater and thermal sensors attached on the OFHC holder to perform precision temperature control for the diamond crystal.

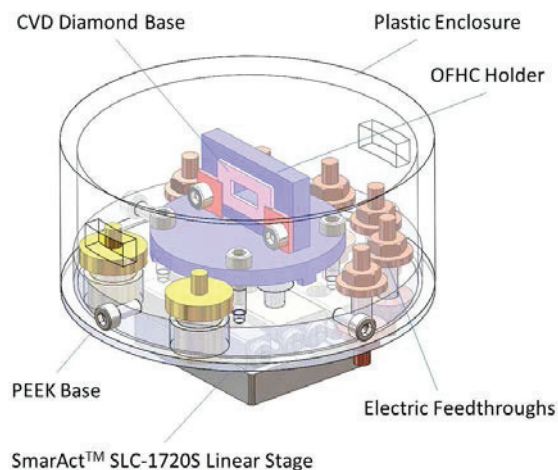


Figure 3. A 3-D model of the diamond crystal holder with plastic enclosure and linear stage.

MODULAR FLEXURE STAGE GROUPS FOR THE DIAMOND CRYSTAL ALIGNMENT

Since the set of four diamond crystal holders has a similar multidimensional nanopositioning requirement for diamond crystal handling alignment, a group of customized modular flexure mechanisms were designed to minimize the design effort with affordable manufacturing cost. As shown in Figure 4, same base rotary stage and tip-tilting stage designs were used for the four sets stage groups. Special adapters are designed to configure the different diamond crystal mounting orientations for each individual diamond crystals C0, C1, C2 and C3. Figure 5 shows a 3-D model of the stage group configured for the diamond crystal C0. There are two flexural positioning stages stacked together to control the diamond crystal's pitch and roll angular adjustment. A pair of commercial Piezo-motor-driven linear stages (SmarAct™ SLC-1720S) are applied to manipulate the diamond crystal along the diamond crystal's optical surface.

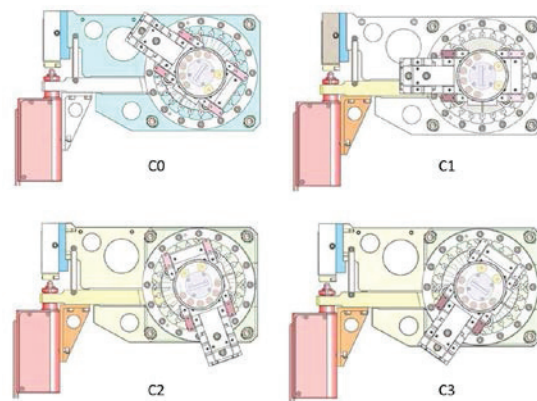


Figure 4. A 3-D model of a group of customized modular flexure mechanisms with different diamond crystal mounting orientations for individual diamond crystals C0, C1, C2 and C3.

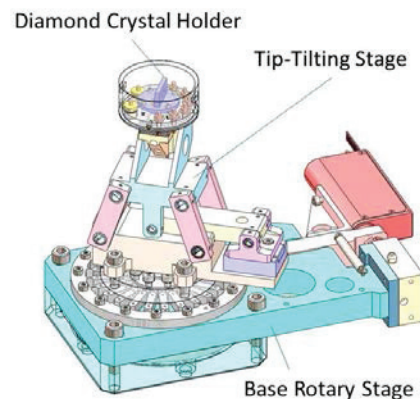


Figure 5. A 3-D model of the stage group configured for the diamond crystal C0.

Design of the Base Rotary Stage

Argonne-developed laminar overconstrained weak-link mechanisms are used to construct the precision base rotary stage for diamond crystal's pitch alignment. Compared with traditional kinematic flexure mechanisms, laminar overconstrained weak-link mechanisms provide much higher structure stiffness and stability. The commercially available stacking thin-metal flexural structure sheets manufactured by photochemical machining processes with lithography technique provide ultrahigh positioning sensitivity and stability [8]. Figure 6 shows a 3-D model of the base flexural rotary stage APS Z8-6301. The structure design is similar to the Z8-73 flexural stage for APS CDDW monochromator [9]. Two linear actuators are mounted on the base plate and sine bar serially to drive the flexural sine bar structure. The rough adjustment is performed by a Newport™ 8310 closed-loop controlled Picomotor™ actuator with a ~30 nm step size. A Physik Instrumente (PI)™ P-753-11C closed-loop-controlled PZT with capacitance sensor provides a 0.1 nm resolution for the fine positioning.

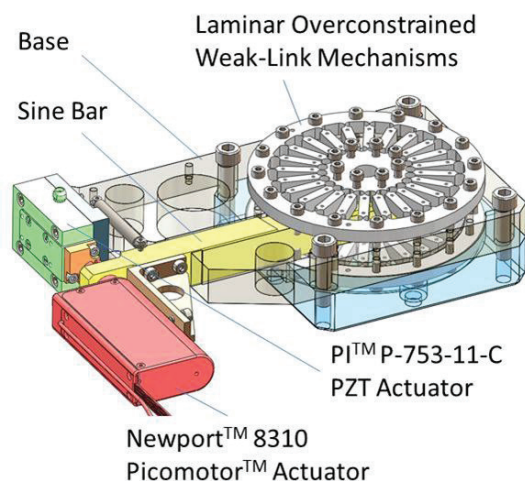


Figure 6. A 3-D model of the base flexural rotary stage APS Z8-6301.

Design of the Tip-Tilting Stage

The flexural tip-tilt stage provides a roll adjustment for the diamond crystal alignment as shown in figure 7. It is constructed with a 4-bar flexural bearing structure operated by a SmarAct™ PZT-driven linear stage SLC-2430 to provide a precise angular positioning around the diamond crystal roll-axis [10]. The adapter plate

between the tip-tilting stage and the base rotary stage is designed to configure the different diamond crystal mounting orientations for each individual diamond crystal.

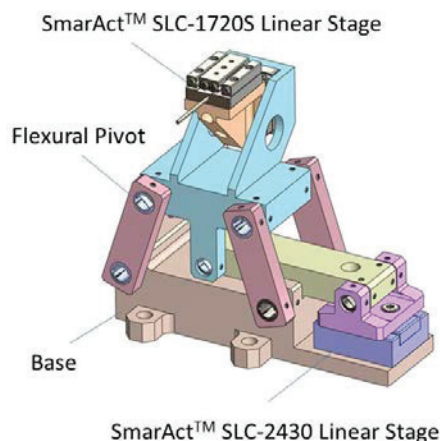


Figure 7. A 3-D model of the flexural tip-tilt stage for the diamond crystal roll alignment.

LINEAR BASE STAGE FOR X-RAY PULSE DELAY-TIME CONTROL

As shown in Figure 2, the flexural stage group for diamond crystal C1 is mounted on a commercial precision linear stage (PI-MICOS UPM-160 from PI™) to perform the x-ray pulse delay time control.

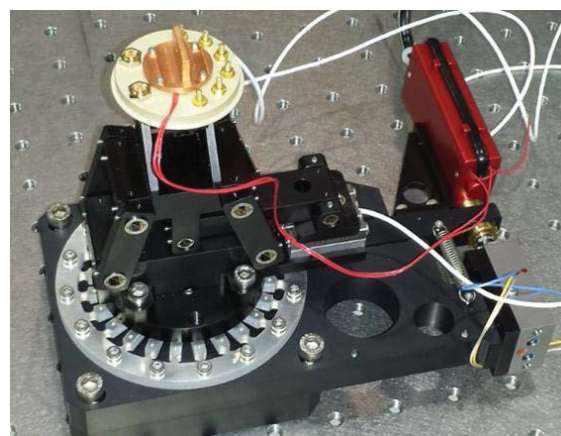


Figure 8. A photograph of the nanopositioning stage group with diamond crystal holder configured as the diamond crystal C0 for the four-crystal x-ray split-and-delay line instrument.

SUMMARY

A prototype of nanopositioning stage group for the four-crystal hard x-ray split-and-delay line instrument has been manufactured and assembled at the APS. It is configured as the

diamond crystal C0 for the four-crystal x-ray split-and-delay line. Figure 8 is a photograph of the nanopositioning stage group with diamond crystal holder. Table 1 summarizes the design specifications of the flexural stage group. Figure 9 shows test results for the stage group's structural stability. The control system of the prototype stage group is integrated with an Experimental Physics and Industrial Control System (EPICS) [11]. Tests of the nanopositioning stage group and diamond crystal holder with x-rays are in progress.

Table 1. Design specifications of the APS Z8-63 two-dimensional flexural stage group.

Z8-6301 base flexural rotary stage	
Overall dimensions (mm)	210 (L) x 148 (W) x 44 (H)
Normal load capacity (kg)	1
Driver type	PZT and Picomotor™
Encoder type	capacitance sensor/optical sensor
Angular travel range (mrad)	18
Min. incremental (nrad)	5
Z8-6302 flexural tip-tilting stage	
Overall dimensions (mm)	104 (L) x 52 (W) x 87 (H)
Normal load capacity (kg)	0.1
Driver type	Piezo-motor-driven linear stage
Encoder type	optical sensor
Angular travel range (mrad)	70
Min. incremental (nrad)	250

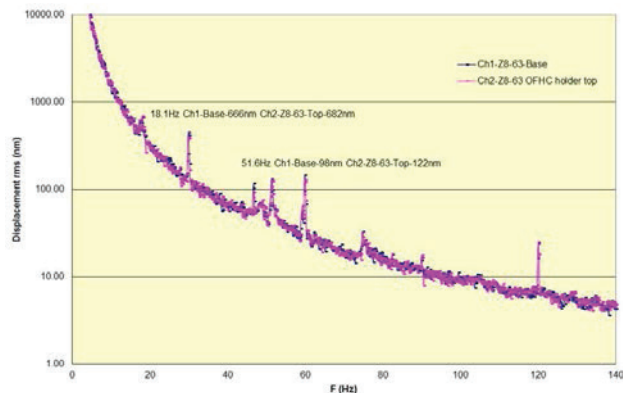


Figure 9. Preliminary vibration test results: data series 2 was measured on the top of the OFHC holder, which showed that the Z8-63 stage group does not amplify the vibration from the mounting base (data series 1). Vibration spectra were measured with compact accelerometers on a noisy table.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] G. Grubel, et al., Nucl. Instrum. Methods Phys. Res. B 262, 357–367 (2007).
- [2] <http://www.aps.anl.gov/Upgrade/APS> Upgrade project.
- [3] Y. P. Stetsko, et al., Appl. Phys. Lett. 103, 173508 (2013).
- [4] Technical Report No. SLAC-R-593, SLAC, 2002.
- [5] W. Roseker, et al., Opt. Lett. 34, 1768 (2009).
- [6] W. Roseker, et al., J. Synchrotron Radiat. 18, 481 (2011).
- [7] S. Stoupin, et al., J. Appl. Cryst. (2014) 47, 1329–1336.
- [8] U.S. Patent granted No. 6,607,840, D. Shu, T. S. Toellner, and E. E. Alp, 2003.
- [9] U.S. Patent granted No. 9,008,272, D. Shu, Y. Shvyd'ko, S. Stoupin, R. Khachatryan, K. Goetze, and T. Roberts, (2015).
- [10] D. Shu, et al., J. Phys. Conf. Ser. 425 052004 (2013).
- [11] <http://www.epics.org>

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FAST CALIBRATION TOOLS AND STRATEGY FOR FIVE-AXIS MILLING MACHINE

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INTRODUCTION

Geometrical accuracy is a key performance criterion for machine tools, with ever increasing requirements pushing machine builders to search for alternatives to keep the pace of the market. The trend to include workpiece measuring and validation functionalities in the machines emphasizes the need for accurate and traceable machines. Accuracy is typically limited by thermal effects (positioning errors dependent on temperature) in the first place, and geometric errors (positioning errors not dependent on temperature) in the second place. In five-axis machines, geometric errors gain relevance due to higher errors in rotary axes [1,2].

Software compensation of geometric errors has long been studied as a cost effective way of improving machine accuracy up to matching its repeatability [1]. A wide variety of calibration methods have been proposed, based on different measurement tools and strategies and compensation models.

The focus of this work is on developing a fast and simple calibration solution for five-axis milling machines of medium to large working volume (a few m³ of working volume). The goal is to enable the machine user to verify and calibrate the machine regularly with minimum machine occupation time, with relatively inexpensive equipment that can be integrated in the machine and operated by a regular machine operator.



FIGURE 1. 5 axis milling machine

The work will be demonstrated on a large milling machine, with three linear axes with moving column and two rotary axes in the spindle head.

STATE OF THE ART

A wide variety of methods have been proposed for measuring the accuracy of machine tools. Direct methods address error components individually. Positioning errors are typically measured with laser interferometers, straightness errors can be measured against straight references, such as taut wires or laser beams, or with laser interferometers and more complex optical paths. Angular errors can be measured with similar methods as straightness errors, but also electronic levels are a popular option. Finally, squareness errors can be measured against an angle standard, such as a granite or ceramic square, but also with laser systems [1]. Direct methods provide good information on individual axis performance, but a full measurement of a machine would require a very long time and much equipment.

Indirect methods involve simultaneous movements of all machine axes and provide an alternative which is typically better when all machine errors need to be measured. These methods are mostly based on physical calibrated artifacts and on laser tracking devices. Physical artifacts with features to be measured in 1D, 2D and 3D have been presented, and the error mapping is based on comparing the calibration of the artifact in a CMM and the measurement of the same features by the machine.

Laser trackers have gained popularity in the last years, mainly for the calibration of large machines, where artifacts face many limitations. Laser trackers measure directly 3D coordinates of a reflector mounted as a tool, providing a fast a convenient measurement. Their accuracy (typically tens of microns) is however not sufficient for high precision machines. This lead to the implementation of multilateration

algorithms, where the workspace is measured sequentially from three or four different positions, improving significantly the calibration accuracy [3].

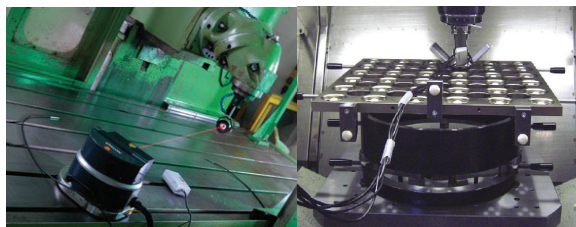


FIGURE 2. (left) ETALON Laser Tracer [3] (right) 3D ball plate artifact [4]

CALIBRATION STRATEGY

An artifact based calibration strategy is proposed here, adapted to fast and simple operation. The type of machines addressed here has a larger workspace than the workspace covered by the two rotations of the spindle head, and thus a sequential approach to error mapping is found most convenient.

In a first step, the workspace of the three linear axes will be mapped. A measuring head is installed on the machine, and it is used to measure the position of several spheres on an artifact. The artifact is placed on different positions in the machine workspace, and the different distances between the spheres measured by the machine are compared with the calibrated distances in order to obtain the 21 parametric errors of a 3 axis machine.

In order to minimize the measuring time, the calibration based on measurements in only two orthogonal planes is proposed here.

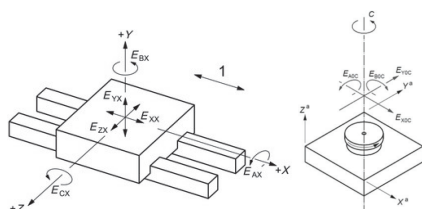


FIGURE 3. (left) errors in linear axis (right) errors in rotary axis

Once the compensation model of the linear axes is implemented, the measurements performed by moving the three linear axes can be considered precise enough, and remaining errors would come from the rotary axes. Taking advantage of this, the calibration of the two axes can be performed in a chase-the-ball style, but

measuring against a fixed sphere and moving the two rotary axes. The measuring head allows for very fast measurement, and therefore a large number of positions can be measured. This allows the identification of more parametric errors that can be obtained with other measuring methods, and therefore a more precise compensation.

CALIBRATION INSTRUMENTS

The successful implementation of this strategy is based on the development of some measurement equipment that ensures high accuracy and fast and automated operation. The main development is a self-centering measuring head, which can measure on two adapted artifacts with precision spheres, one for calibration of linear axes and the other for rotary axes. The artifacts are presented first.

3 linear axes calibration artifact

“1D ball array” has been selected as the most suitable artifact for machine tool calibration, considering accuracy, usability, cost and possibility of automation and machine integration. This artifact contains 11 precision spheres evenly distributed over a length of 1500mm, thus with 150mm distance between them. As imposed by the measuring head that is presented later, the spheres are made of ferromagnetic material, stainless steel in this case. The spheres are of precision grade G25, which requires roundness error below 0.6 μm , for a diameter of 20mm.

The main requirement for its use as calibration reference is the stability of the relative positions between the spheres. The position of the spheres is calibrated on a CMM, and the measurements on the machine are compared to the CMM measurements, and therefore, this should not change with time, temperature or position and orientation in the machine.

A carbon fiber reinforced plastic (CFRP) tube has been chosen as structural material for the artifact, given the low weight, high stiffness and thermal stability it provides. The spheres are rigidly mounted onto the carbon fiber tube. The coefficient of thermal expansion of the tube is of 1 $\mu\text{m}/(\text{m K})$ (approx. ten times lower than steel or cast iron), and thus for a 1.5m distance between spheres, the relative distance between them changes at most 1.5 $\mu\text{m}/\text{K}$. Since the artifact changes very little with temperature, the effect of the temperature changes on the machine will be

seen in the machine calibration, and it will be considered as part of the calibration uncertainty.

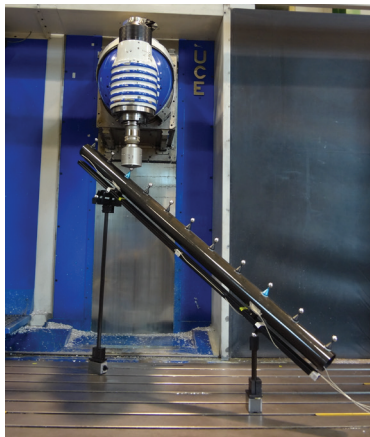


FIGURE 3. CFRP artifact in machine

In order to minimize the deformation of the artifact when it is mounted on the machine, the tube is always supported on two fixed points, located on the airy points of the tube. This means that the support points are symmetrically arranged around the center of the tube of length L and are separated by a distance d , defined by:

$$d = L/\sqrt{3} = 1.6/\sqrt{3} = 0.923 \text{ m}$$

2 rotary axes calibration artifact

The calibration of the rotary axes comes after the compensation of the errors of the linear axes. Thanks to this, the calibration can be done by measuring against a fixed sphere, and covering the whole workspace of the rotary axes. A tool for aligning the sphere with the rotary axis of the spindle is needed.

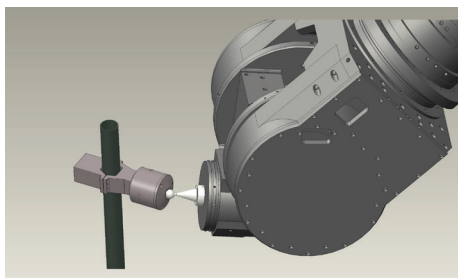


FIGURE 4. Setup for angular axis tests

Measuring head

Calibrated artifacts for the error mapping of CMMs and machine tools have been long used. One important limitation for its extensive use is the long time it takes to perform all necessary

measurements. Standard touch probes are used, which require to contact the sphere in several points, making it a slow process. Contact force can also produce displacements on the artifact, which would add uncertainty to the calibration.

Measuring heads with three sensors arranged orthogonally have been developed, both based on contacting and non-contacting sensors, which can locate the center of the sphere in a single step. The machine is moved to a position where the sensors are in range, and the reading of the three sensors is process to find the machine coordinates where the sphere would be centered relative to the head.

In this case, the accuracy of the measurement depends on the accuracy of the measuring head. The three sensors need to accurately measure the distance to the sphere, and their position and orientation needs to be known accurately for the transformation of coordinates. In this work, an evolution of this measuring principle is presented which adds a self-centering functionality. The accuracy of the measurement only relies then on the repeatability of the sensors and on an approximate knowledge of their orientation.

Configuration and self-centering principle

The measuring head proposed in this work contains three displacement sensors that measure the distance from the sensor to a target sphere. Non-contacting inductive sensors are used, which give an analogue voltage signal related to the distance to a ferromagnetic target.

The main innovation with this measuring head is the implementation of a self-centering function [5] that reduces the accuracy requirements on the measuring head without reducing the accuracy of the calibration. This is based on an iterative algorithm that moves the machine following the reading from the measuring head, bringing it to a nominal relative position between the sphere and the head.

Following this procedure, the position of the center of the sphere is taken from the machine axes positions. Therefore the accuracy of the measuring head does not influence the calibration, only the repeatability of the sensors in finding the nominal position with respect to the sphere. This allows for using simpler and cheaper displacement sensors, since the

electronics for linearizing their output is not needed.

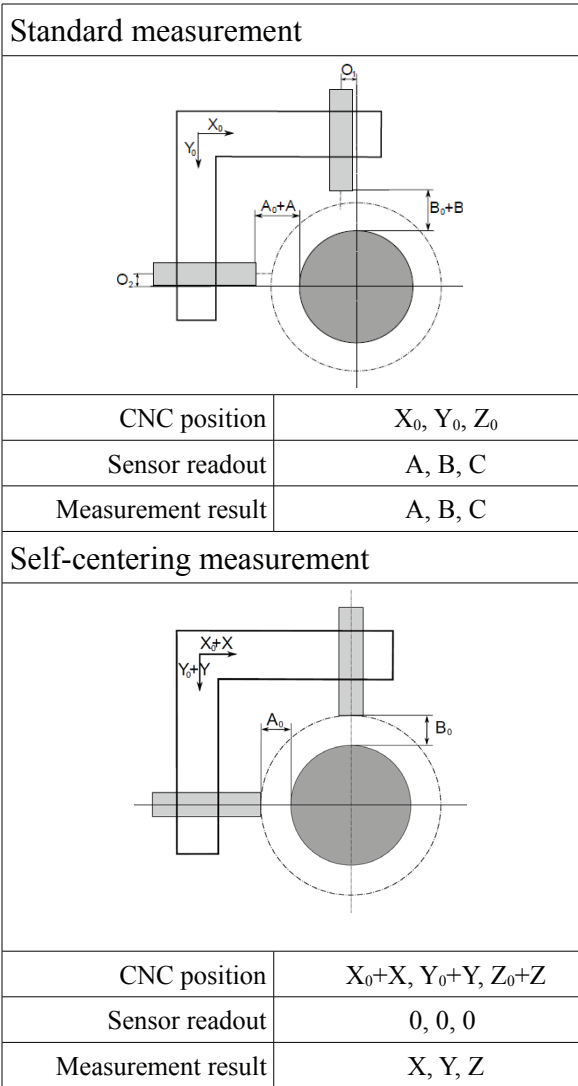


FIGURE 5 Measurement strategies

Measuring head prototype

A prototype measuring head has been designed and built which implements the functionality described above. It is designed to be mounted in the spindle as the application in this work requires, but other arrangements are foreseen for other applications. Its main characteristics are the following:

- 3 non-contacting displacement sensors with 1mm measuring range arranged orthogonally for measuring against 20mm spheres.

- Wireless communication with a PC for measurement data transmission. Possible to implement directly on CNC.
- Battery operated, fully wireless operation.

The prototype used in this work is shown next:



FIGURE 6. Measuring head prototype

Control implementation

The control implementation of the self-centering functionality is summarized in Figure 7. There is a central control unit which governs all the calibration test. This is currently an external PC but could be integrated in the machine control if necessary. This control unit communicates with the measuring head, receiving continuously the reading from the three displacement sensors. The control unit is also in continuous communication with the machine control, reading axes positions. The PC is also able to send movement commands to the machine. Thanks to this, all the tests shown in this work are fully commanded from the external PC. A dedicated software has been developed which handles all the communication between the devices, runs all measurements and process the results.

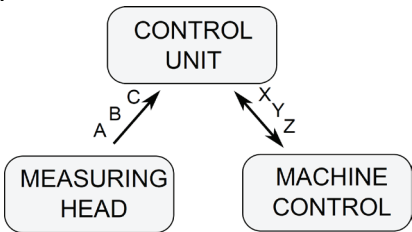


Fig. 7: (A) Measuring head control implementation

The self-centering algorithm is implemented in the control unit. It follows the iterative diagram shown in Figure 10.

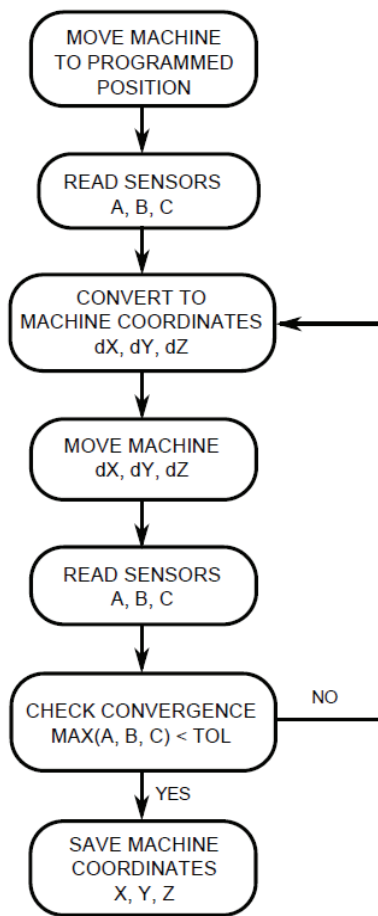


FIGURE 8. Self-Centering algorithm

Self-centering repeatability

The repeatability in reaching the nominal position around a sphere when the self-centering function is activated is the best performance measure for the measuring head, and self-centering function, and it indicates the uncertainty that it adds to the machine calibration system. In this test, starting from the nominal position, random displacements are applied on the machine, within the range of the sensors, and the centering function is activated afterwards, which should lead the machine back to its nominal position. The variation in the machine coordinates to which the centering function converges is shown in Figure 9. High repeatability in the centering function is observed.

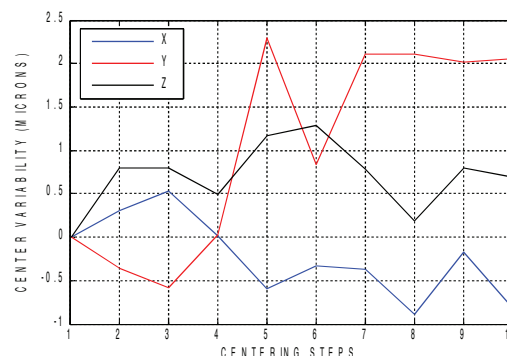


FIGURE 9. Self-Centering repeatability

CONCLUSIONS AND FUTURE WORK

A new solution for five axis machine tool error mapping has been presented. Classical artifact based indirect calibration with 1D ball array has been selected as optimal solution for industrial application. The main novelty relies on the use of a low cost self centering measuring head and algorithm with micron range repeatability, and thus its validity for this application.

Future work involves the practical implementation of the error mapping and compensation strategy on the machine.

REFERENCES

- [1] Schwenke H, Knapp W, Haitjema H, Weckenmann A, Schmitt R, Delbressine F. Geometric error measurement and compensation of machines—An update. *CIRP Ann - Manuf Technol* 2008;57:660–75.
- [2] Bringmann B, Knapp W. Model-based “Chase-the-Ball” Calibration of a 5-Axes Machining Center. *CIRP Ann - Manuf Technol* 2006;55:531–4. doi:10.1016/S0007-8506(07)60475-2.
- [3] Schwenke, H., M. Franke, J. Hannaford, and H. Kunzmann. 2005. “Error Mapping of CMMs and Machine Tools by a Single Tracking Interferometer.” *CIRP Annals - Manufacturing Technology* 54 (1): 475–78.
- [4] Bringmann B, Knapp W. Machine tool calibration: Geometric test uncertainty depends on machine tool performance. *Precis Eng* 2009;33:524–9.
- [5] European Patent Application N.15380021.4

PRECISION MANUFACTURING OF THERMOPLASTIC ELASTOMER MICRO RINGS

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INTRODUCTION

The result of the current trend towards product miniaturization is a demand for advances in micro-manufacturing technologies and their integration in new manufacturing platforms.

Micro injection molding (μ IM) is one of the most suitable micro manufacturing processes for flexible mass-production of multi-material functional micro components. The μ IM can deliver sub-100 mg down to few mg polymer components, with achievable dimensional tolerances in micrometer range. However, in order to achieve such process capabilities, μ IM poses great challenges in terms of quality control of both the manufacturing process and the products itself. Shot sizes down to 100 mg and even lower than 10 mg, dimensional tolerances in the micrometer range and surface roughness in the sub- μ m range call for process control systems and metrology solution with very low relative uncertainties (<0.1 - 1%) [1].

The mass-replication nature of the process calls for fast monitoring of process parameters and product geometrical characteristics. In this direction, the present study addresses the possibility to develop a micro manufacturing platform for micro assembly injection molding with real-time process/product monitoring and metrology. The study represent a new concept yet to be developed with great potential for high precision mass-manufacturing of highly functional 3D multi-material (i.e. including metal/soft polymer) micro components.

Towards a fully automated phono cartridges production (see *FIGURE 1a*), the current study focus on the optimization of the thermoplastic elastomer (TPE) suspension ring (see *FIGURE 1b*, *1c*) micro injection molding production, as key components for final product functionalities. Specifically, the optimization study is focused on establishing the correct metrology strategy to be later implemented as program routine to control quality of the produced micro rubber rings. The study proves the importance of using tool

geometries as reference calibrated artefacts to establish the correct process windows for optimal part quality within tolerance limits in the micrometer dimensional range.

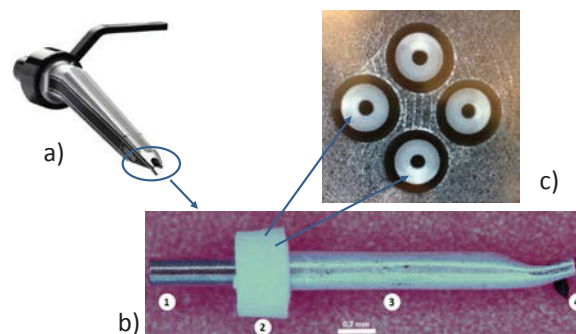


FIGURE 1: a) high performance DJ cartridge, b) (b-1) magnet (b-2) TPE suspension ring (b-3) Aluminum cantilever (b-4) Diamond tip; c) 2D visual inspection of produced suspension rings.

CASE STUDY – PHONO CARTRIDGES

Micro assembly of three-dimensional miniaturized components is of difficult realization due to the reduced dimensions, and hence mass, of the components, and the raising of adhesion forces due to scale effects. The required positioning in the μ m/sub- μ m precision is also challenging to achieve if top-down approaches when downscaling mechanical assembly systems are applied. In order to overcome these limitations, the ability of polymers to flow around pre-assembled and positioned miniaturized inserts in a micro mold cavity can be exploited by the implementation of micro insert molding [2]. To achieve this final goal, a the first μ IM mold prototype that replaced the time consuming through-thickness hot embossing process was designed with the aim of realizing a high throughput production of micro rings with no preassembled magnet and aluminum cantilever (figure b-1,b-3) inside the mold during injection. The developed mold,

figure 2, consists of a three-plate system with a moving middle plate. Two pneumatic pistons push the middle plate forward on each side when the mold is being opened. When the mold closes the ejection side is pushed against the middle plate, where four tapered guides ensures consistent alignment. The closing forces provided by the ejection side overcomes opening pressure from the pneumatic pistons and the middle plate mates with the injection side. The suspension ring has an outside target diameter of $1500 \pm 10 \mu\text{m}$, inside target diameter of $450 \pm 20 \mu\text{m}$. When assembled it holds in place the preassembled aluminum cantilever, magnet and diamond tip as seen in Figure 1b.

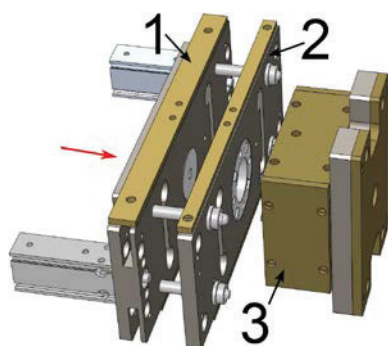


FIGURE 2: Exploded mold view. Red arrow indicates Injection side, 1: Injection plate, 2: moving middle plate and 3: ejection side plate.

MICRO INJECTION MOLDING

To evaluate the influence of process parameters on the replication of the inner and outer ring diameters, a statistically designed set of experiments was carried out. The micro injection moldings were executed on Witmann-Battenfeld Micro Power 15 with an injection unit consisting of an $\varnothing 14 \text{ mm}$ screw for plasticization and metering combined with a separate $\varnothing 5 \text{ mm}$ injection plunger. After screening experiments on which different process parameters were varied to find the optimal process window a designed set of experiment (DOE) was performed. A full general factorial design ($2 \times 3^2 = 18$ experiments) was run.

TABLE1: DOE process parameters.

Process parameters	Values
Melt Temperature [$^{\circ}\text{C}$]	210, 220
Injection Speed [mm/s]	60, 80, 100
Packing pressure [bar]	300, 400, 500

Generally, process parameters were set to optimize the resulting quality of the produced geometries considering machine capability, material physical properties and process feasibility. In this respect, three process parameters were identified, after screening experiments, as the most significant affecting final parts quality. Different levels of the melt were chosen in accordance with the material supplier specifications, as a balanced value to avoid material degradation and premature solidification during the filling of the cavity. Injection speeds were selected to enhance replication, bearing in mind the possibility to yield, with a too high speed, high internal stresses, air entrapment and flashes. Finally, packing pressure different levels were selected as trade-off between acceptable shrinkage compensations, dimensional accuracy automatic and effective part demolding.

For each experimental process setting within the designed set of experiments 50 parts were produced to obtain part quality repeatability due to machine stability. After process stabilization 50 parts were collected for each experiment and only 5 parts randomly picked and measured.

MEASUREMNT STRATEGY

Dimensional quality control of the produced polymer parts was carried out using a focus variation microscope. The instrument selection allowed for off-line part quality detection, figure 3a, after suspension rings were produced. Future development foresee the same image elaboration technique for in line capabilities. The suspension ring were placed with the gate pointing up in order to secure a plane contact surface on the movable instrument stage. For each suspension ring three measurements repetitions to quantify the polymer rings inner and outer diameter were performed as described in Figure 3b.

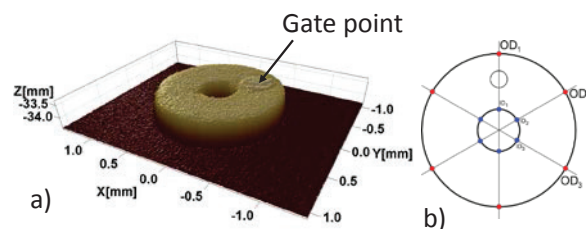


FIGURE 3: a) 3D captured image of TPE suspension ring; b) measurement strategy adapted to dimensionally characterize produced rubber rings.

A DeMeet 220 Optical Coordinate measuring machine was employed to measure/calibrate mold cavities (outer suspension ring diameter) and pin diameter (inner suspension ring diameter).

Measuring uncertainty was calculated following ISO 15530-3 [3]. The expanded standard uncertainty was calculated as:

$$U = k \cdot \sqrt{u_{cal}^2 + u_p^2 + u_w^2 + u_{wt}^2 + u_b^2}$$

Where U = Expanded uncertainty including confidence level. k = Coverage factor of 2 for a confidence level of 95,45%. u_{cal} = Standard uncertainty associated with uncertainty of the calibration of the calibrated workpiece stated in the calibration certificate (Type B evaluation). u_p = Standard uncertainty associated with the measurements procedure, repeated of mold cavities were used to give a more reliable results that reflect the actual measurement procedure (Type A evaluation). u_w = Standard uncertainty of workpiece form error, estimated as max - min of repeated measurements inner and outer mold diameter (Type B evaluation). u_{wt} = Standard uncertainty associated with the variation of workpiece due to thermal expansion which is considered negligible as described in [4]. u_b = Standard uncertainty associated with systematic error from calibrated reference circle (Type B evaluation).

TABLE 2: Uncertainty contributions (unit: μm)

Parameter	ID	OD
Std. Unc. Calibration: u_{cal}	0,3	0,3
Std. Unc. Measureing procedure: u_p	0,71	0,83
std. Unc. Systematic error: u_b	0,16	0,05
Std. Unc. Workpiece (form error): u_w	0,77	1,18
Expanded Uncertainty (95% (k=2)): U	2,2	2,9

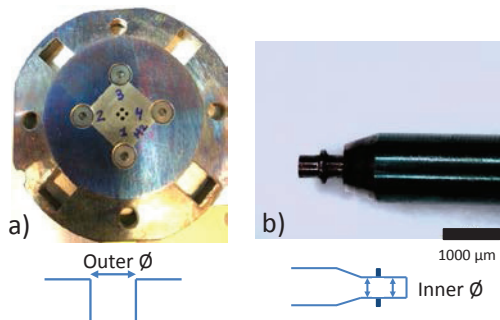


FIGURE 4: a) Mold plate containing 4 cavities; b) pin entering the cavity during injection molding production of the suspension ring.

DISCUSSION AND RESULTS

Inner and outer diameter of the produced rubber rings of each DOE experimental setting were

measured using an infinite focus variation microscope enabling the 3D reconstruction of all the fabricated parts. Dimensional measurements analysis of outer diameter (OD) and inner diameter (ID) are shown in the form of main effect plot in figure 5 and figure 6. Error bars in figure 5 and 6 indicate experimental measurement uncertainty calculated including instrument calibration, measurements repeatability and process reproducibility.

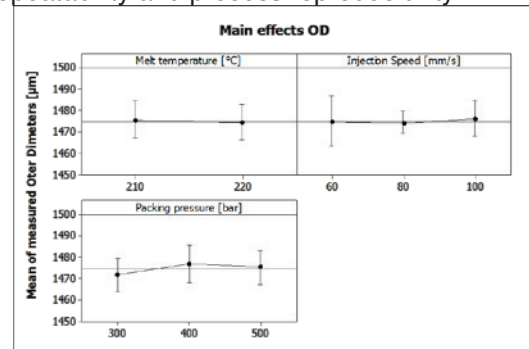


FIGURE 5: Main effect plot of measured suspension rings Outer Diameters.

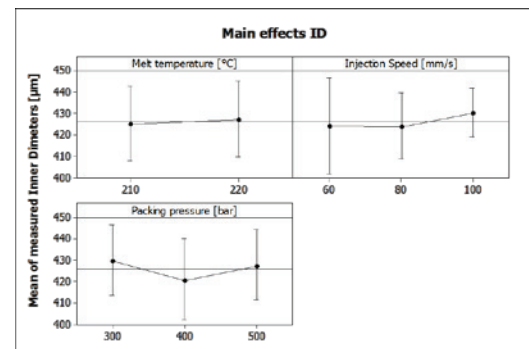


FIGURE 6: Main effect plot of measured suspension rings Inner Diameters.

Significant effects cannot be observed within the tested process windows due to the large measured products variation. Nevertheless, measurements on ring's outer diameter (figure 5) have shown considerably less product variation compared with internal diameters measured on the equivalent suspension rings. Injection speed was the factor mostly affecting the product variation. Lower level of injection speed (60 mm/s) produces the largest internal diameter variations with extreme values within $\pm 20 \mu\text{m}$. Also, in correspondence to the lower level of injection speed the outer diameter largest variation ($\pm 12 \mu\text{m}$) was measured. Parts quality conformance within tolerance limits was carried out using the measurements results from the experimental DOE. Effects of the different

process settings (18 DOE experiments) quantified for the produced TPE suspension rings were compared with target design dimensions (i.e. mold technical drawings) and calibrated measurements carried out on the mold, figure 7 and 8. Both histograms showed that if ID and OD of the produced parts within the investigated process windows are compared with target design dimensions, part conformance assessment could lead to wrong results.

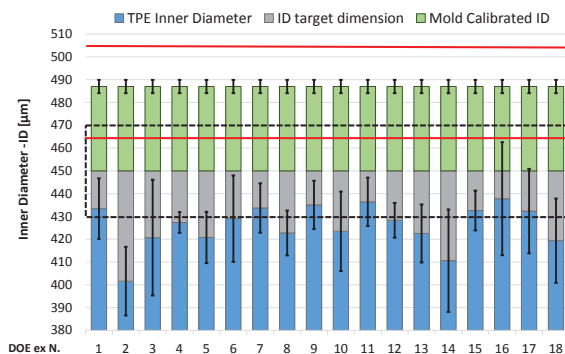


FIGURE 7: Quality conformance assessment of TPE micro rings. Error bars on TPE ID indicate experimental measuring uncertainty. Error bars on the calibrated ID pins dimension indicate expanded uncertainty $U=2,2 \mu\text{m}$.

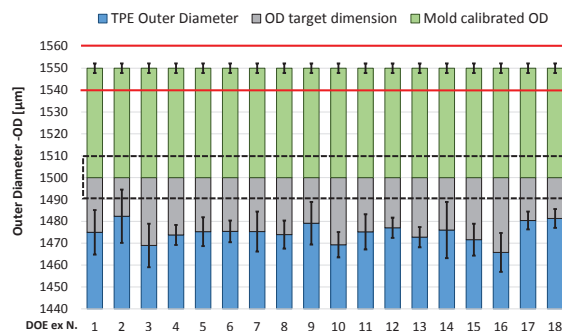


FIGURE 8: Quality conformance assessment of TPE micro rings. Error bars on TPE OD indicate experimental measuring uncertainty. Error bars on the calibrated OD mold cavities dimension indicate expanded uncertainty $U=2,9 \mu\text{m}$.

In fact, in both diagram the dotted rectangular areas indicate the theoretical tolerance limits of ID and OD ($\pm 10 \mu\text{m}$ and $\pm 20 \mu\text{m}$ respectively) which lead to the conclusion that parts are produced close to the optimal process window yielding part quality within tolerance limits. Even more, looking at this lower section of the histograms wrong conclusion could be made if polymer shrinkage behavior was investigated for

future part dimensioning related to shrinkage compensation. Finally, comparing the produced suspension rings (ID and OD) with the real mold features dimension, upper section of the histograms, it is possible to observe that almost none of the experimental process setting produce part that are within the tolerance range (red lines) specified by the manufacture to provide the required functionalities.

CONCLUSIONS

In the present study an approach for micro injection molded TPE micro rings production optimization has been investigated. A design of Experiment was run to identify the most influential process parameters affecting product part quality. Injection speed of 60 mm/s was identified as the parameter that drastically increases the product dimensional variation. Maximum dimensional variation of $\pm 20 \mu\text{m}$ was quantified for suspension rings ID. Moreover, based on experimental setup, the present study demonstrate the importance of calibrating master feature in the produced mold to ensure correct product quality acceptance within product tolerance limit.

ACKNOWLEDGEMENTS

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REFERENCES

- [1] Hansen H.N., et. al. Dimensional Micro and Nano Metrology, CIRP Annals Manufacturing Technology. 2006 55(2):721-743.
- [2] Michaeli W, et.al. Micro assembly injection moulding Microsyst Technol. 2006, 12: 616- 619.
- [3] International Organization for Standardization (ISO). Geometrical product specifications (GPS) - coordinate measuring machines (CMM): Technique for determining the uncertainty of measurement - part 3: Use of calibrated workpieces or measurement standards (ISO 15530-3:2011), 2011.
- [4] Tosello G., Hansen H.N., Gasparin S.et. al., Applications of dimensional micro metrology to the product and process quality control in manufacturing of precision polymer micro components, CIRP Annals, 2009, 58:467-472.

A STUDY OF FACTORS AFFECTING MATERIAL REMOVAL IN COMPUTER-CONTROLLED BONNET POLISHING

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ABSTRACT

Computer-controlled Bonnet Polishing (CCBP) with multi-axis machining is an enabling technology which is capable of fabricating ultra-precision surfaces with sub-micrometer form accuracy and surface roughness in the nanometer range. With the increasing demands for functional enhancement, it is of paramount importance to understand how various polishing process parameters affect the ability to accurately produce a superfinished surface with geometrically dimensional shapes in the submicrometer order. As a result, this paper presents an experimental investigation of the factors affecting the material removal in bonnet polishing. The optimal process parameters and the significance of the important parameters in CCBP are identified. The results shed some light on the optimization of the CCBP.

Keywords: Bonnet Polishing, Material Removal, ultra-precision Machining, Optimization

INTRODUCTION

To meet the optical surface requirements for a wide variety of freeform optics, ultra-precision freeform surfaces are usually fabricated by ultra-precision machining technologies with sub-micrometer form accuracy and nanometric surface finish. [1] Computer-controlled Bonnet Polishing (CCBP) originally invented at London's Optical Sciences Laboratory and developed by Zeeko Ltd has been exploited for commercial production of the robotic polishing system [2]. It possesses the advantage of high polishing efficiency, mathematically tractable influence function, and flexibly controllable spot size with variable tool hardness [3].

As a result, computer-controlled bonnet polishing (CCBP) is an enabling technology which is capable of fabricating ultra-precision freeform surfaces with sub-micrometer form accuracy and surface roughness in the nanometer range, especially for difficult-to-

machine and ferrous materials, which are not amenable using other ultra-precision machining technologies such as single-point diamond turning and raster milling [4].

With the increasing demands for functional enhancement, it is of paramount importance to have a better understanding on how various polishing process parameters affecting the ability in polishing a superfinished surface with geometrically dimensional shapes in sub-micrometer level. The materials removal analysis is clearly a fundamental element in achieving the corrective polishing and optimization of the polishing process.

Nowadays, the acquisition of tool influence functions still depends largely on the experience and skill of the machine operator through an expensive trial-and-error approach when new materials, new surface designs or new machine tools are used. To further optimize and better understand the polishing process, the Taguchi method is used to design the experiments to investigate the optimal process parameters and the significance of the important parameters in CCBP.

The experiments are divided into three parts, i.e. Part A, Part B and Part C. In Part A, an investigation was conducted on the optimization of parameters affecting material removal rate in polishing stainless steel. Part B aims at optimizing parameters that affect the surface roughness in polishing a superhard materials such as reaction bonded silicon carbide (RB SiC). In Part C, the effect of slurry on surface quality is investigated.

EXPERIMENTAL DESIGN

In both Part A and Part B, a Zeeko IRP 200 seven axis ultra-precision freeform polishing machine was used to conduct the experiments of the present study (see *FIGURE 1*). The polishing bonnet is assembled on the main

spindle, while the sample is clamped on the C axis. The polished samples were measured by the Zygo Nexview 3D Optical Surface Profiler as shown in FIGURE 2.

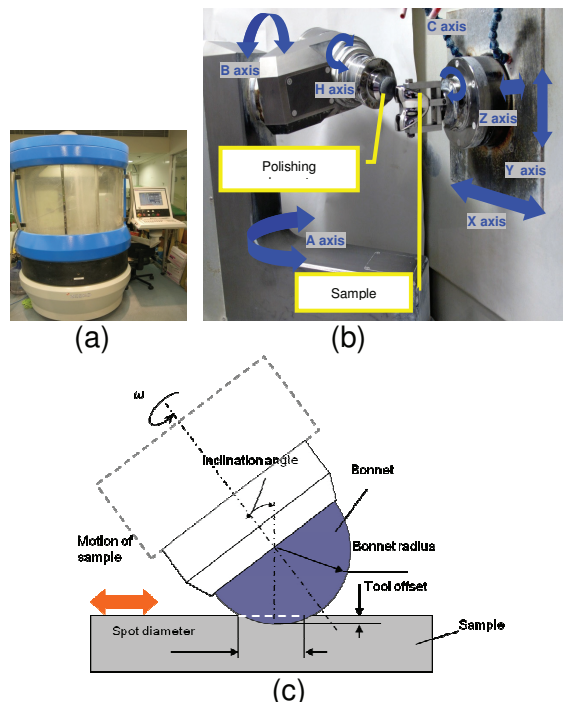


FIGURE 1. (a) Zeeko IRP200 Ultra-precision freeform polishing machine and (b) Configuration of the multi-axis ultra-precision polishing machine for conducting mechanical polishing [1] (c) Process parameters of CCBP.



FIGURE 2. Zygo Nexview 3D Optical Surface Profiler.

Part A. The optimization of parameters affecting material removal rate in polishing stainless steel by CCBP

A Taguchi Approach was used to design experiments. There are more than ten parameters in the CCBP process, which can be mainly divided into 2 groups: (1) process parameters, such as spindle speed and feed

rate, and (2) material parameters, for instance slurry and polishing cloth material.

A series of experiments were conducted using five fixed process parameters and six control factors (each with three levels), as shown in TABLE 1 and TABLE 2. Hence, the Taguchi design method was configured using a Double Orthogonal Array, in which the six control factors were arranged in an L18 (3^6) Orthogonal Array as shown in TABLE 3.

TABLE 1 The fixed process parameters of Part A

Fixed factors	Levels
Tool radius	20 mm
Polishing cloth	PU LP-66
Machining time	4 min
Workpiece material	S136
Particle material	Al ₂ O ₃

TABLE 2 The control factors and levels of Part A

No	Control factors	Levels		
		1	2	3
A	Head speed (rpm)	500	1000	1500
B	Inclination angle (°)	5	15	25
C	Tool pressure (bar)	0.5	1	1.5
D	Tool offset (mm)	0.12	0.24	0.36
E	Concentration	1:9	1:6	1:3
F	Particle Size (μm)	3.22	6.69	9.41

TABLE 3 L18 (3^6) Orthogonal Array

Exp. No.	Control factors					
	A	B	C	D	E	F
1	1	1	1	1	1	1
2	2	2	2	2	2	2
3	3	3	3	3	3	3
4	1	1	2	2	3	3
5	2	2	3	3	1	1
6	3	3	1	1	2	2
7	1	2	1	3	2	3
8	2	3	2	1	3	1
9	3	1	3	2	1	2
10	1	3	3	2	2	1
11	2	1	1	3	3	2
12	3	2	2	1	1	3
13	1	2	3	1	3	2
14	2	3	1	2	1	3
15	3	1	2	3	2	1
16	1	3	2	3	1	2
17	2	1	3	1	2	3
18	3	2	1	2	3	1

Part B. The optimization of parameters affecting surface roughness in polishing RB SiC by CCBP
RB SiC is well-known to be a superhard difficult-to-machine material. In Part B, 2mm x 2mm RB SiC plates were polished. Four factors were chosen to be investigated as shown in TABLE 4 and TABLE 5 lists the fixed parameters, and thus a L9 Orthogonal Array was used to design the experiment. The fixed factors are listed in TABLE 6.

TABLE 4 The control factors and levels of Part B

No	Control factors	Levels		
		1	2	3
A	Head speed (rpm)	500	1000	1500
B	Inclination angle (°)	5	15	25
C	Tool pressure (bar)	0.5	1	1.5
D	Tool offset (mm)	0.12	0.24	0.36

TABLE 5 The fixed process parameters of Part B

Fixed factors	
Tool radius	20 mm
Feed rate	200 mm/min
Pitch	0.5 mm
Polishing cloth	PU LP-66
Sample material	RB SiC
Particle material	Silicon carbide
Concentration	1:10
Particle Size	3.0±0.4µm

TABLE 6 L9 (3²) Orthogonal Array

Exp. No	Factor			
	A	B	C	D
1	1	1	1	1
2	1	2	2	2
3	1	3	3	3
4	2	1	2	3
5	2	2	3	1
6	2	3	1	2
7	3	1	3	2
8	3	2	1	3
9	3	3	2	1

Part C. The investigation of effect of slurry material in polishing RB SiC by CCBP

In Part C, a preliminary investigation of the effect of polishing powder in the CCBP process of superhard material was conducted. The optimized polishing parameters in the last section were applied to the polishing process of reaction bonded silicon carbide plate. Two 2mm

x 2mm RB SiC were polished for 15 minutes and the volume of material removed was measured by the Zygo NexView every 3 minutes. The detailed experimental setup is summarized in Table 7.

TABLE 7 Experimental setup of Part C

Sample No.	Polishing powder	Particle size	Polishing parameters
1	Aluminum oxide (Al ₂ O ₃)	3.0 ±0.4µm	Head speed: 1500 rpm; Inclination angle: 25°; Tool pressure: 0.5 bar; Tool offset: 0.36 mm
2	Silicon Carbide (SiC)	3.0 ±0.4µm	

RESULTS AND DISCUSSION

In the experiments of Part A, a higher-the-better S/N ratio was used and a larger S/N ratio infers that the corresponding factor level setting provides a larger material removal rate in bonnet polishing. The results show that the *inclination* angle, head speed, particle diameter, tool offset, slurry concentration and tool pressure are the significant factors in a declining sequence affecting the material removal rate, and the combination of the optimal factor level for the material removal rate is found to be A3B3C1D3E3F3 in the present study.

In Part B, a smaller-the-better S/N ratio was used to analyse and find out the optimal set of parameters to polishing RB SiC. RB SiC is a very difficult to machine material. Some sets of parameters have almost no effect reducing surface roughness. The optimal set of parameter is A3B2C1D2. A verification test was then carried out to examine the optimized parameters. There was about 20% improvement of the surface roughness.

The experimental results are plotted and as shown in FIGURE 3 and FIGURE 4. With the use of Al₂O₃ polishing powder, the volume of material removed and the material removal rate are higher than that of SiC powder in the first 3 minutes.

The material removal rate of Al₂O₃ polishing powder decreases with time and this leads to insignificant increase of the volume of material removed. SiC polishing powder is able to maintain a stable material removal rate and a steady increase of volume of material removed.

This may be due to the wear of Al_2O_3 polishing powder in the polishing process as the hardness of Al_2O_3 is lower than that of RB SiC. Although it is different from steel, with the appropriate polishing parameters, CCBP is able to polish RB SiC even with relative softer polishing powder.

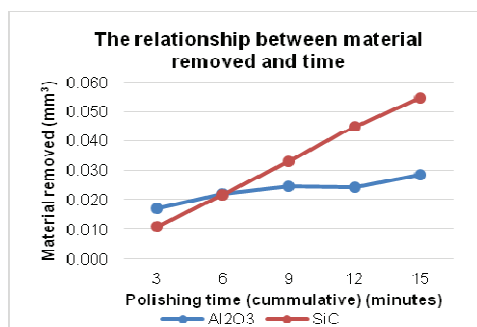


FIGURE 3. The relationship between material removed and time

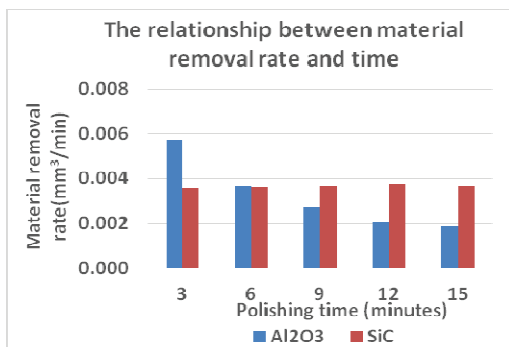


FIGURE 4. The relationship between material removal rate and time

CONCLUSIONS

The Taguchi design experiments are used to identify the optimal operational parameters and the significance of the important parameters for the volumetric material rate and surface roughness in CCBP. The Taguchi trials show that the inclination angle, head speed, particle diameter, tool offset, slurry concentration and tool pressure are the significant factors in a declining sequence affecting the material removal rate.

The results of the Taguchi design process indicate that the optimal process parameters in present experiments include the head speed of 1500 rpm, the inclination angle of 25°, the tool pressure of 0.5 bar, the tool offset of 0.36 mm, the concentration of 1:3, and particle diameter of 9.41 μm , when the S136 sample material was polished using a bonnet with a radius of 20 mm

and LP-66 polishing cloth, pure water additives, and Al_2O_3 abrasive particles.

For the improvement of surface finishing of RB SiC, the optimal set of parameters regarding only the factors studied in the present study is head speed of 1500 rpm, inclination angle of 15°, tool pressure of 0.5 bar and tool offset of 0.24mm. The results not only provide an important mean for a better understanding of the factors affecting materials removal but also help to optimize the material removal in CCBP. It is also found that with appropriate polishing slurry, the material removal rate can be maintained at a constant rate at a relatively longer period of time.

ACKNOWLEDGEMENT

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REFERENCES

- [1] Cheung CF, Ho LT, Charlton P, Kong LB, To S, Lee WB. Analysis of surface generation in the ultraprecision polishing of freeform surfaces, Proc. I Mech Eng B-J Eng. 2010; 224(1): 59-73.
- [2] Walker DD, Brooks D, Freeman R, King A, McCavana G, Morton R, Riley D, Simms J. The first aspheric form and texture results from a production machine embodying the Precession Process, Proc. SPIE, Optical Manufacturing and Testing IV, 267. 2001; 4451: 268.
- [3] Bingham RG, Walker DD, Kim DH, Brooks D, Freeman R, Riley D. Novel automated process for aspheric surfaces, Proc. SPIE, Current Developments in Lens Design and Optical Systems Engineering. 2000; 4093: 445-450.
- [4] Cheung CF, Kong LB, Ho LT, To S. Modelling and simulation of structure surface generation using computer controlled ultra-precision polishing, Precis Eng. 2011; 35: 574-590.

MILLING BIFURCATION AT EXTENDED AXIAL DEPTHS OF CUT

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INTRODUCTION

In science and engineering fields, new discoveries are typically followed by a burst of follow-on research activity and corresponding publications. These discoveries tend to serve as a catalyst to the research community and often result in new insights, improved understanding of fundamental phenomena, and enhanced modeling capabilities. For machining, one such period of rapid progress began in the mid-19th century [1-3]. During this time, self-excited vibrations were first described using time-delay differential equations. The notion of “regeneration of waviness” was promoted as the feedback mechanism (time-delay term), where the previously cut surface combined with the instantaneous vibration state dictates the current chip thickness, force level, and corresponding vibration response. This work resulted in analytical algorithms that were used to produce the now well-known stability lobe diagram that separates the spindle speed-chip width domain into regions of stable and unstable behavior [4]; see Fig. 1.

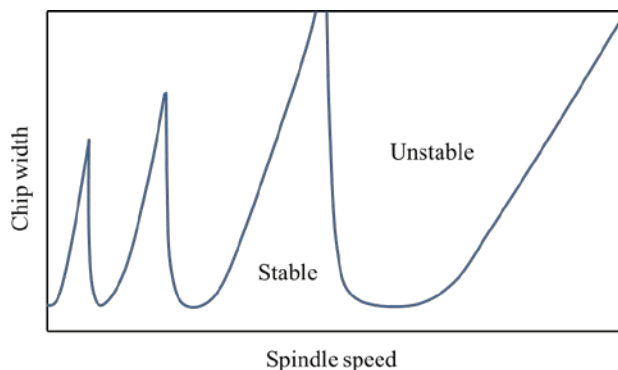


Figure 1. Example stability lobe diagram.

In 1998, a similar step forward in the understanding of machining behavior was realized. Davies *et al.* used once-per-revolution sampling to characterize the synchronicity of cutting tool motions (measured using a pair of orthogonal capacitance probes) with the tool rotation in milling [5]. This approach was an experimental modification of the Poincaré maps used to study state space orbits in nonlinear dynamics. They observed the traditional quasi-periodic chatter associated with the secondary (subcritical) Hopf, or Neimark-Sacker, bifurcation that can occur for systems described by periodic time-

delay differential equations. This was an expected result and was observed as an elliptical cluster of once-per-revolution sampled points in the x-y measurement plane perpendicular to the endmill axis. This elliptical collection of points occurred because the chatter frequency was incommensurate with the tooth passing frequency and quasi-periodic behavior was obtained. However, they also recorded period-3 tool motion (i.e., motion that repeated with a period of three cutter revolutions) during partial radial immersion milling. This period-3 motion manifested itself as three distinct clusters of once-per-revolution sampled points in the x-y plane. They noted that this behavior was “inconsistent with existing theory” [5].

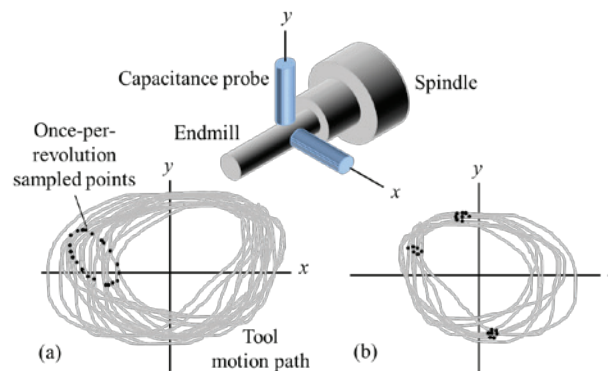


Figure 2. Once-per-revolution sampling of cutting tool motions (a) Hopf instability; (b) period-3 instability.

In this paper, bifurcation diagrams are used to study milling at extended axial depths of cut (beyond the stability limit). Initial results show that unexpected behavior occurs in the unstable region and that new milling strategies may be applied based on these results.

BIFURCATION DIAGRAMS

A bifurcation diagram enables the evolution of system behavior (e.g., tool motion) with a control variable of interest (such as axial depth of cut in milling) to be efficiently observed. The diagram uses the periodic sampling strategy to identify periodic (or aperiodic) responses over the selected range of the control variable. For milling, the tool motion in the feed, x, or y direction is sampled once per spindle revolution for

a given axial depth of cut (and fixed spindle speed). This produces a sequence of points over multiple cutter revolutions (see Fig. 2 for example). This collection of points is then truncated to remove the transient portion of the motion (typically the first few milliseconds).

For stable milling with motion that is periodic with the cutting force (i.e., only forced vibrations are present), these sampled points repeat each revolution because the cutting force and subsequent vibration response is periodic with the spindle rotation. The superposition of all these repeated points therefore gives a single point (or nearly so) on a bifurcation diagram of axial depth (horizontal axis) versus once-per-revolution sampled tool motion (vertical axis).

For a higher axial depth at the same spindle speed, secondary Hopf instability may occur and then the motion is quasi-periodic with tool rotation because the chatter frequency is (generally) incommensurate with the tooth passing frequency. In this case, the once-per-revolution sampled points do not repeat and they form a distribution (as shown in Fig. 2a). When plotted on the bifurcation diagram, this distribution appears as a vertical “spread” of points.

For period-2 instability, on the other hand, the motion repeats only once every other cycle (i.e., it is a sub-harmonic of the forcing frequency). In this case, the once-per-revolution sampled points alternate between two solutions. On the bifurcation diagram, the points appear in two distinct vertical locations (recall that the vertical axis is the sampled tool motion). For period- n instability, the sampled points appear at n vertical locations. The bifurcation diagram construction from results at multiple axial depths of cut for a selected spindle speed is depicted in Fig. 3.

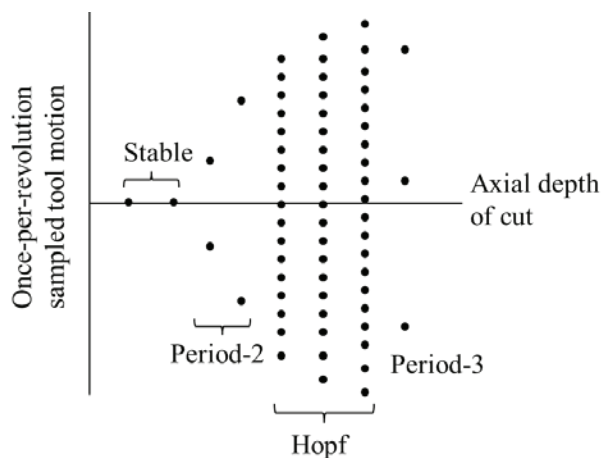


Figure 3. Description of stable/unstable behavior for a milling bifurcation diagram.

TIME-DOMAIN SIMULATION

Time-domain simulation entails the numerical solution of the governing equations of motion for milling in small time steps. It is well-suited to incorporating all the intricacies of milling dynamics, including the nonlinearity that occurs if the tooth leaves the cut due to large amplitude vibrations and complicated tool geometries (including runout, or different radii, of the cutter teeth, non-proportional teeth spacing, and variable helix). The simulation is based on the Regenerative Force, Dynamic Deflection Model described by Smith and Tlustý [6]. As opposed to stability lobe diagrams that provide a “global” picture of the stability behavior, time-domain simulation provides information regarding the “local” cutting force and vibration behavior (at the expense of computational efficiency) for the selected cutting conditions. The simulation proceeds as follows:

1. the instantaneous chip thickness is determined using the vibration of the current and previous teeth at the selected tooth angle
2. the cutting force is calculated
3. the force is used to find the new displacements
4. the tooth angle is incremented and the process is repeated. Modal parameters are used to describe the system dynamics in the x (feed) and y directions, where multiple degrees of freedom in each direction can be accommodated.

The instantaneous chip thickness depends on the nominal, tooth angle-dependent chip thickness, the current vibration in the direction normal to the surface, and the vibration of previous teeth at the same angle. The chip thickness can be expressed using the circular tool path approximation as $h(t) = f_t \sin \phi + n(t - \tau) - n(t)$, where f_t is the commanded feed per tooth, ϕ is the tooth angle, n is the normal direction, and τ is the tooth period. The tooth period is defined as $\tau = \frac{60}{\Omega N_t}$ (sec), where Ω is the spindle speed in rpm and N_t is the number of teeth. The vibration in the direction of the surface normal for the current tooth depends on the x and y vibrations as well as the tooth angle according to $n = x \sin \phi - y \cos \phi$.

For the simulation, the strategy is to divide the angle of the cut into a discrete number of steps. At each small time step, dt , the cutter angle is incremented by the corresponding small angle, $d\phi$. This approach enables convenient computation of the chip thickness for each simulation step because: 1) the possible teeth orientations are predefined; and 2) the surface created by the previous teeth at each angle may be stored. The cutter rotation $\phi = \frac{360}{SR}$ (deg)

depends on the selection of the number of steps per revolution, SR . The corresponding time step is $dt = \frac{60}{SR \Omega}$ (sec). A vector of angles is defined to represent the potential orientations of the teeth as the cutter is rotated through one revolution of the circular tool path, $\phi = [0, d\phi, 2d\phi, 3d\phi, \dots, (SR - 1)d\phi]$. The locations of the teeth within the cut are then defined by referencing entries in this vector.

In order to accommodate the helix angle for the tool's cutting edges, the tool may be sectioned into a number of axial slices. Each slice is treated as an individual straight tooth endmill, where the thickness of each slice is a small fraction, db , of the axial depth of cut, b . Each slice incorporates a distance delay $r\chi = db \tan \gamma$ relative to the prior slice (nearer the cutter free end), which becomes the angular delay between slices: $\chi = \frac{db \tan \gamma}{r} = \frac{2db \tan \gamma}{d}$ (rad) for the rotating endmill, where d is the endmill diameter and γ is the helix angle. In order to ensure that the angles for each axial slice match the predefined tooth angles, the delay angle between slices is $\chi = d\phi$. This places a constraint on the db value. By substituting $d\phi$ for χ and rearranging, the required slice width is $db = \frac{d \cdot d\phi}{2 \tan \gamma}$. Using the time-domain simulation approach, the forces and displacements may be calculated. These results are then once-per-revolution sampled to generate the bifurcation diagrams.

RESULTS

As noted, the semi-discretization method was applied to predict Hopf and period-2 bifurcation. These predictions were verified experimentally and were reported in [7]. The up milling tests were completed using an 8 mm diameter endmill (mounted in a shrink fit tool holder) with one cutting edge, a 45 deg helix angle, and a 96 mm overhang. The radial depth of cut was 0.4 mm to provide highly interrupted cutting conditions. The aluminum workpiece and tool combination yielded a force model with a specific cutting force of 644 MPa and 69.7 deg force angle.

The long, slender tool exhibited a single dominant bending mode. The modal parameters for the x (feed) and y directions are provided in Table 1.

Table 1: Modal parameters obtained from impact testing.

Direction	Natural frequency (Hz)	Damping ratio (-)	Stiffness (N/m)
x	721	0.009	4.1×10^5
y	721	0.009	4.1×10^5

The stability diagram obtained from the semi-discretization method is plotted in Fig. 4. Lines are added that depict the spindle speeds that were used for the extended milling bifurcation diagrams. The modifier "extended" is used to emphasize that axial depths well beyond the predicted stability limit were used.

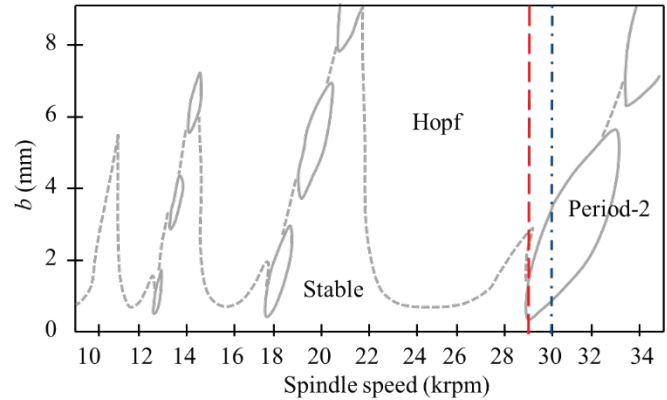


Figure 4. Stability lobe diagram with stable and unstable (Hopf and period-2) zones (from [7]). Test speeds for the bifurcation diagrams are identified: (dashed line) 29000 rpm; (dash-dot line) 30000 rpm.

In Fig. 4 it is observed that the 29000 rpm (dashed) line predicts stable behavior up to an axial depth of 0.3 mm. Period-2 instability then occurs up to an axial depth of 2 mm. Stable performance is again predicted until 2.5 mm. Hopf instability then occurs for higher depths of cut.

The corresponding bifurcation diagram is presented in Fig. 5. The vertical axis represents the once-per-revolution sampled x-direction tool motions, while the horizontal axis is the axial depth of cut. The transition from stable to period-2 motion occurs at 0.26 mm; the period-2 instability persists with increasing amplitude until an axial depth of 2.1 mm. Stable operation is again obtained at 2.1 mm and is maintained until 2.4 mm. Hopf instability then occurs. The second stable zone validates the closed islands of stability depicted in Figs. 3 and 5. Regions of period-7 (3.38 mm to 3.54 mm) and period-5 (4.48 mm to 4.7 mm) instability are also observed. This behavior is not predicted by existing milling stability

theory and, to the authors' knowledge, has not been previously presented in the literature.

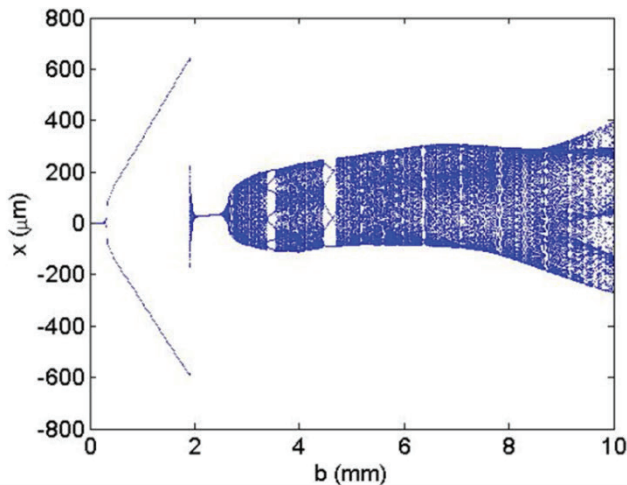


Figure 5. Extended milling bifurcation diagram (29000 rpm).

A bifurcation diagram was also produced for 30000 rpm (dash-dot line in Fig. 4). In Fig. 6, a transition from stable to period-2 motion is seen at 0.68 mm. The period-2 behavior persists to an axial depth of $b = 1.26$ mm where combination period-2 and quasi-periodic motion occurs. This is exhibited by the two separate vertical spreads in points. At 1.88 mm, the motion changes to quasi-periodic only and a single, vertical distribution of once-per-revolution sampled points is seen.

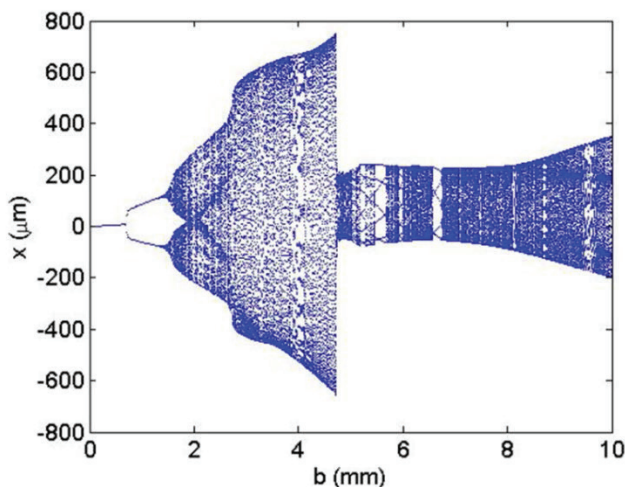


Figure 6. Extended milling bifurcation diagram (30000 rpm).

The significance of the extended range bifurcation diagram is seen at an axial depth of 4.76 mm. A

dramatic amplitude reduction is observed at this depth, even though the cut remains unstable. This reduced amplitude at a high axial depth could provide acceptable cutting conditions, while offering a high material removal rate. Additionally, period-7 instability is predicted in the range from $b = 5.48$ mm to 5.66 mm. As noted, milling stability theory does not predict this behavior. These results call for additional analysis and experiments to better understand the predicted behavior.

These results demonstrate that the extended milling bifurcation diagram is a powerful tool to enable a detailed view of unstable behavior and elicit an improved understanding of milling dynamics. It is expected that this new understanding will lead to new milling strategies that improve productivity.

REFERENCES

- [1] Arnold RN (1946) The mechanism of tool vibration in the cutting of steel. Proceedings of the Institute of Mechanical Engineers, 154.
- [2] Tobias SA, Fishwick W (1958) The chatter of lathe tools under orthogonal cutting conditions. Transactions of the ASME, 80:1079-1088.
- [3] Tlustý J, Poláček M (1963) The stability of machine tools against self-excited vibrations in machining. Proceedings of the ASME International Research in Production Engineering Conference, Pittsburgh, PA, 465-474.
- [4] Schmitz T, Smith S (2009) Machining Dynamics: Frequency Response to Improved Productivity. Springer, New York, NY.
- [5] Davies MA, Dutterer BS, Pratt JR, Schaut AJ (1998) On the dynamics of high-speed milling with long, slender endmills. Annals of the CIRP 47(1):55-60.
- [6] Smith, K.S. and Tlustý, J., 1991, An overview of modeling and simulation of the milling process, Journal of Engineering for Industry, 113: 169-175.
- [7] Govekar, E., Gradišek, J., Kalveram, M., Insperger, T., Weinert, K., Stepan, G., and Grabec, I., 2005, On stability and dynamics of milling at small radial immersion, Annals of the CIRP, 54/1: 357-362.

Categorization and Review of Existing Micro-mirror Array Technologies

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ABSTRACT

The aim of this paper is to categorize and compare the performance of existing two-dimensional (2D) micro-mirror array (MMA) devices and to establish physical bounds on the performance of such technologies. Existing MMA technologies are categorized according to their actuation approach and key examples are discussed to demonstrate how each category achieves their specific combinations of performance capabilities. Performance plots are provided that compare a variety of mirror-array-capability metrics such as mechanical half-range, maximum acceleration or stepping rate, and energy density. Considerations of mirror size, array size, and fill-factor are also addressed. The performance of an MMA design created by the authors will be included within these plots to highlight its unique advanced capabilities.

MICRO-MIRROR REVIEW EFFORT

The published performance capabilities of existing micro-mirror designs have been studied and compared for the purposes of this paper. Figure 1 provides a plot of the number of micro-mirror design publications per year since their conception.

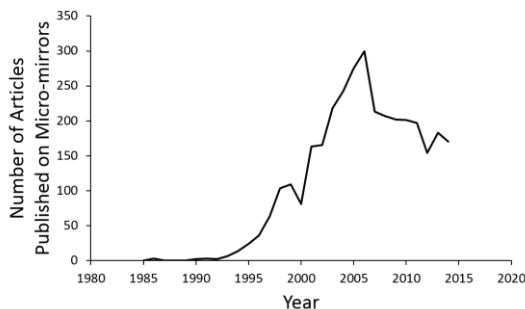


FIGURE 1. MMA publications over time

In this paper, we restrain our focus on 2D MMA designs that achieve at least tip and tilt DOFs. 87 designs from 22 companies and 21 academic

research groups have been categorized and compared. The reported performance specs of each design have been adapted to a form that can be compared universally for all mirror designs. The performance of other micro-mirror designs will be discussed in a later paper.

MICRO-MIRROR ARRAY CATEGORIZATION

We primarily categorize 2D MMAs according to how they are actuated. We further differentiate these arrays by categorizing them according to whether their mirror surfaces are discrete or continuous (i.e., whether each mirror in the array is an independent well-defined unit or whether each mirror is ambiguously defined as a small portion of one continuous surface that is deformed by a multiplicity of actuators). We have thus divided the designs into eight categories: Plate Discrete, Comb Discrete, Thermal Discrete, Lorentz Discrete, Piezo Discrete, Plate Continuous, Piezo Continuous, and Lorentz Continuous.

1) Plate Discrete Actuation

Electrostatic plate actuators are most commonly used within discrete MMAs because they are compact, fast, require low power consumption, and are easily integrated within MEMS devices. More than 35 designs utilize electrostatic plate actuators from among the complete body of discrete 2D MMA designs that we considered in this study accounting for more than 60% of the designs.

Tip-Tilt-Piston Deformable Mirror (TTP-DM) [1], [2], developed by Boston Micromachines in 2007, consists of hexagonal mirrors with 3 DOFs (tip, tilt, and piston). Figure 2 illustrates the architecture of TTP-DM device. Each mirror is actuated by three independent pairs of electrostatic plates from below enabling 6 mrad (0.35°) of tip or tilt motion and 2μm of piston stroke. The high fill-factor (99%), fast time

¹These are both first authors as their contributions to this paper are equal

response (17us), and high resolution (14 bits) make TTP-DM ideal for applications in wavefront control. TTP-DM's relatively small range of motion, however, limits its ability to steer light.

Transparent Networks Inc. (currently defunct) developed an MMA with $\pm 10^\circ$ tip and tilt range capability [3, 4]. Figure 3a is a cutaway of the device showing the layered structure. Each mirror connects to a mechanical angular amplifier which is actuated by electrostatic electrodes (Fig. 3b). The mechanical linkage is designed so that the rotation of the mirror is amplified by a factor of 4. The fill-factor is about 75%, but, in principle, this metric could be improved.

Other Plate Discrete 2D MMA designs of note can be found in [6-8] (by commercial companies) and [9-12] (by academic research groups).

2) Comb Discrete Actuation

Comb actuators are another form of capacitive actuation that utilize electrostatic force. The interdigitated teeth of the comb allow higher energy density and larger actuation forces than capacitive parallel plate actuators of the same size. However, the complex geometry of comb actuators increases the design and fabrication complexity of MMAs and thus makes their realization more challenging. About 25% of all discrete 2D MMA designs considered for this paper are classified as Comb Discrete designs.

Alcatel-Lucent developed a 2D tip-tilt-piston MMA actuated by electrostatic combs [13]. Figure 4a shows the mirror structure. The mirror surface is connected to four arms by flexible joints. Each arm is attached to a rotational comb (Fig. 4b). By controlling the voltage on each comb drive, the mirror surface can achieve up to $\pm 4^\circ$ tip/tilt and 5um upward piston motion within 20us time. The reported performance of this design is one of the best among all the 2D MMA devices.

IW Jung et al. [14] from Stanford University developed a comb-actuated MMA design that achieves up to $\pm 0.9^\circ$ tilt, $\pm 0.1^\circ$ tip, 0.07um piston, 99% fill-factor and ~ 10 us response time. Figure 5a shows a top view and Fig. 5b shows a bottom view of the gimbaled structure of the mirror. Although not yet demonstrated, IW Jung et al. report that high fill-factor ($>94\%$), fast response (<100 us), and large rotational angle ($>\pm 10^\circ$) are

feasible for designs with appropriately scaled actuators.

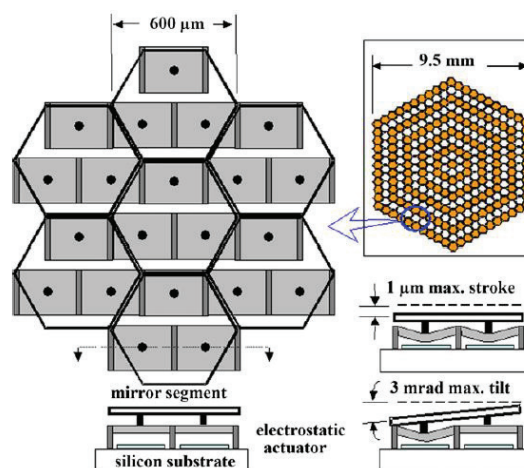


FIGURE 2. Top and side view of TTP-DM

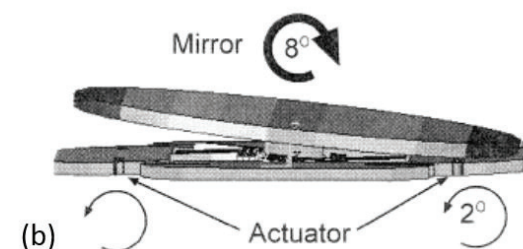
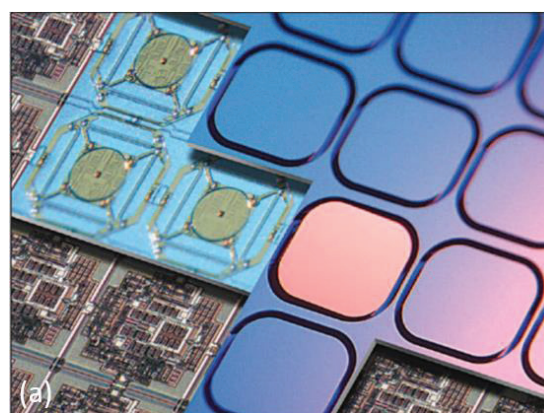


FIGURE 3. Transparent Networks MMA

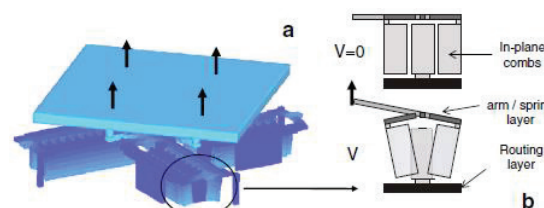


FIGURE 4. Alcatel-Lucent micro-mirror design

We have also created a Comb Discrete 2D MMA design [15, 16]. The predicted performance capabilities of our array are discussed and

compared in Performance Plots Section. Other Comb Discrete MMA designs of note can be found in [17, 18] (by commercial companies) and [19-21] (by academic research groups).

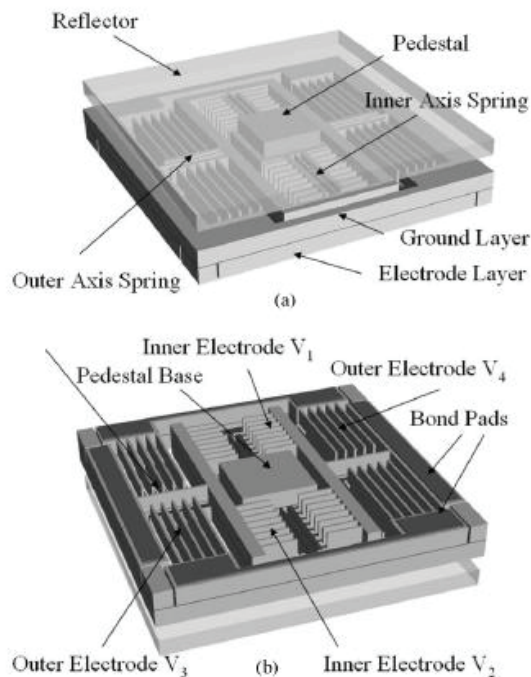


FIGURE 5. Gimbaled micro-mirror design by IW Jung et al.

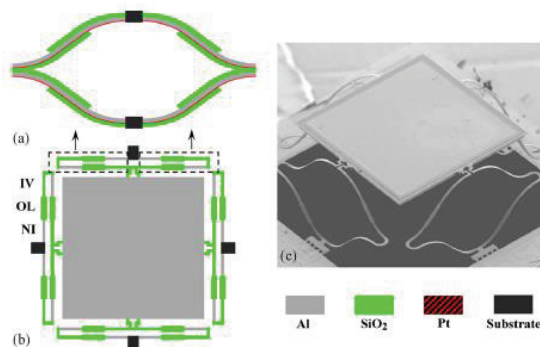


FIGURE 6. Micro-mirror by H. Xie et al.

3) Thermal Discrete Actuation

Electrothermal bimorphs are the most commonly used thermal actuators. Limited by the actuation dynamics, thermally actuated micro-mirrors can only operate at relatively low speed (<100Hz stepping rate), thus less chosen for fast and precise application.

A Thermal Discrete 2D MMA was developed by H. Xie et al. [22] from University of Florida. Figure 6 shows the single mirror design. The mirror surface is suspended by four pairs of

electrothermal bimorph actuators. The MMA has an 88% fill-factor, $\pm 15^\circ$ tip/tilt range and $\sim 310\mu\text{m}$ piston range. The response time of the device is on the order of 10ms.

Other Thermal Discrete 2D MMA designs of note can be found in [23] and [24].

4) Lorentz Discrete Actuation

Electromagnetic actuators utilize Lorentz force generated by electric current in magnetic field. The driving torque is controlled by the strength and direction of the current flow. The main challenge in the design of Lorentz Discrete MMAs is current path planning. In addition, Joule losses and disturbance in magnetic field may be issues in practical applications.

Integrated Micro Machines developed a TTP MMA actuated by Lorentz force (Fig. 7a) [25]. The mirror has a 98% fill-factor and can be driven up to $\pm 6^\circ$ tip/tilt and $\pm 50\mu\text{m}$ piston. Position sensors are integrated in the each mirror (Fig. 7b), and a 1kHz closed-loop servo control is achieved. Due to the actuator design, the single mirror size is relatively large ($3\times 3\text{mm}^2$ and above) comparing to most MMA devices.

Other Lorentz Discrete 2D MMA designs of note can be found in [26] and [27].

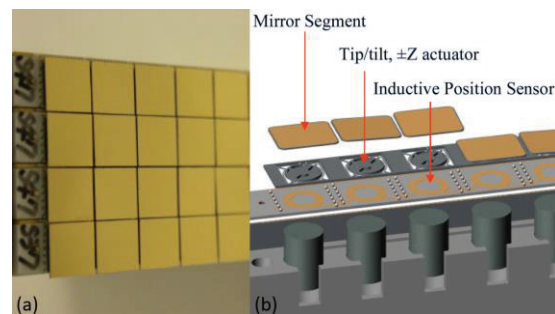


FIGURE 7. Integrated Micro Machines MMA

5) Piezo Discrete Actuation

Piezo actuators are designed to utilize electric-field-induced strain of piezoelectric materials. Typically, piezoelectric materials have large load capacity but small deformation range. Therefore, piezo actuators of discrete 2D MMAs are most commonly in the form of unimorph or bimorph.

There are a few piezo-actuated single micro-mirror designs, but so far there is only one reported 2D MMA with piezoelectric actuation. This MMA is developed by Y. Yee et al. [28] from LG Electronics Institute of Technology. The

2D TT micro-mirror array is actuated by PZT unimorphs (Fig. 8), achieving $\pm 0.7^\circ$ rotation around outer axis and $\pm 0.5^\circ$ around inner axis.

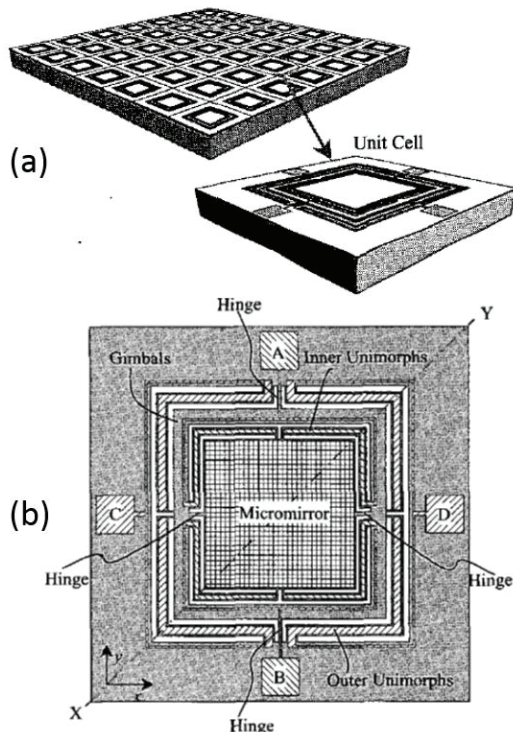


FIGURE 8. MMA Design by Y. Yee et al.

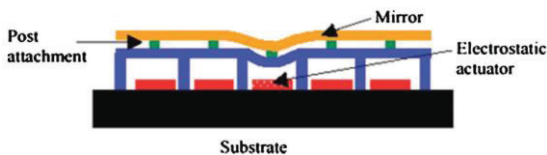


FIGURE 9. Boston Micromachines 4K-DM

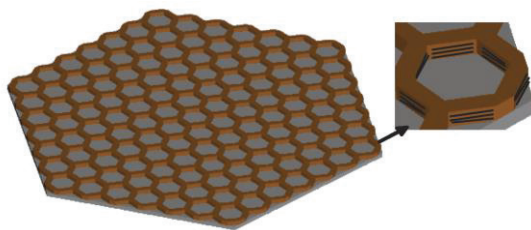


FIGURE 10. Surface Parallel Array DM

6) Plate Continuous Actuation

Boston Micromachines developed a few continuous deformable mirror devices based on same actuator design as TTP-DM [29]. One of them is studied and tested in [30]. Figure 9 shows the basic working principle of this so-called 4K-DM device. It achieves up to 4 μ m of localized deformation within 100 μ s response

time. The electrostatic actuators are placed under the mirror surface with a 400 μ m pitch.

IW Jung et al. [31] developed a single-crystal-silicon continuous membrane deformable mirror. The mirror is capable of ~ 125 nm localized deformation with a pitch of 200 μ m at a fast speed (~ 25 kHz).

7) Piezo Continuous Actuation

AOA Xinetics developed a few continuous deformable mirror devices with piezoelectric actuation [32]. In most of their deformable mirror devices, PZT stack actuators are placed directly under the mirror to generate large deformation force on the mirror surface. One exception is Surface Parallel Array, which is actuated by electrostrictive ceramic bimorph actuator array (Fig. 10). The pitch of the actuators ranges from 1mm to 5mm, corresponding to stroke of 0.5 μ m to 4 μ m. The bandwidth varies between 2.5kHz and 5kHz.

8) Lorentz Continuous Actuation

There is only one company named Imagine Optic that developed continuous deformable mirror devices with Lorentz actuation [33]. A typical Imagine Optic deformable mirror achieves ± 50 μ m piston stroke with ~ 2 mm actuator pitch and ~ 0.5 kHz bandwidth.

PERFORMANCE PLOTS

All 2D MMA designs that achieve at least tip and tilt DOFs from among those studies are plotted in Figs. 11 and 12. Figure 11a provides information about the MMA's dynamic capabilities by showing the maximum angular acceleration against tip/tilt range. The maximum angular acceleration dictates how fast the mirror can be driven. In addition, Fig 11b provides mirror size information by making the scatter dot size proportional to the single mirror size. Figure 12 shows the energy against tip/tilt range.

The green hexagon shown in the figures represents the performance capabilities of a Comb Discrete MMA design created by the authors. This design possesses a transmission feature that enables it to be tuned in such a way that it can be made to slide along the constant energy angled dashed lines in Fig. 11. By comparison, the MMAs with similar or better dynamic capabilities have much smaller single mirror size. The dots corresponding to these MMAs are too small to be seen in Fig. 11b. The thick horizontal dashed line in Fig. 12 represents

the theoretical bound on energy density and the thin horizontal dashed line in the same figure represents the practical bound on achievable energy density. The predicted energy density of our design is very close to the practical bound.

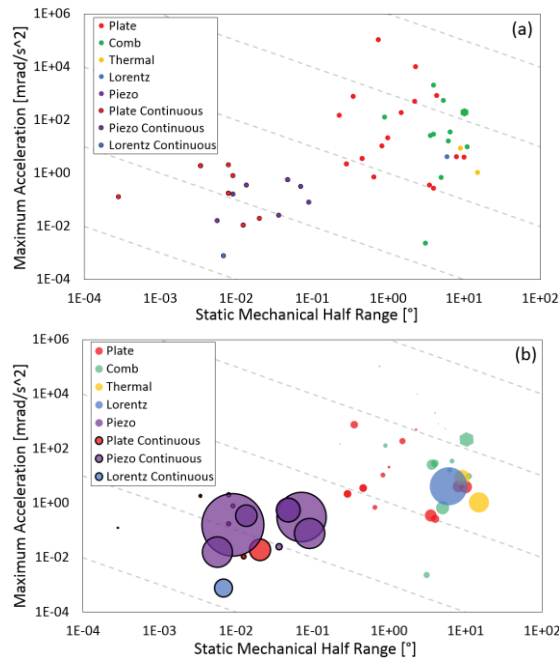


FIGURE 11. Maximum Acceleration vs. Range

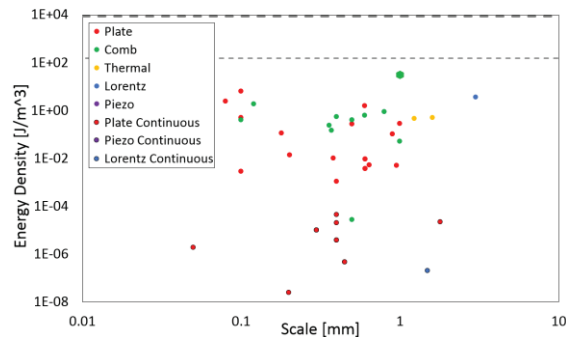


FIGURE 12. Energy Density vs. Size

CONCLUSIONS

The majority of published micro-mirror designs have been studied, categorized, and compared. Plots that capture their performance capabilities have been generated and theoretical bounds have been calculated to help designers recognize how much design improvement is feasible.

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REFERENCES

- [1] Stewart JB, Bifano TG, Cornelissen S, Bierden P, Levine BM, Cook T. Design and development of a 331-segment tip-tilt-piston mirror array for space-based adaptive optics. *Sensors And Actuators A: Physical*:230–8.
- [2] Tip-Tilt-Piston (TTP)-DM. Boston Micromachines. <http://www.bostonmicromachines.com/ttp-dm.htm> (accessed August 2015).
- [3] Bryzek J, Flannery A, Skurnik D. Integrating microelectromechanical systems with integrated circuits. *IEEE Instrum Meas Mag* 2004;7.
- [4] Bryzek J, Nasiri S, Flannery A, Kwon H, Novack M, Marx D, et al. Very Large Scale Integration of MOEMS Mirrors, MEMS Angular Amplifiers and High-voltage, High-density IC Electronics for Photonic Switching. *Nanotechnol. Conf. Trade Show*, vol. 1, 2003, p. 300–3.
- [5] Yu H, Chen H. Development of a novel micromirror based on surface micromachining technology. *Sensors Actuators, A Phys* 2006;125:458–62.
- [6] Helmbrecht M a, He M, Kempf CJ. Scaling of the Iris AO segmented MEMS DM to larger arrays. *Mems Adapt Opt Iii* 2009;7209:1–5.
- [7] Fernandez A. Modular MEMS design and fabrication for an 80 x 80 transparent optical cross-connect switch. *Proc SPIE* 2004;5604:208–17.
- [8] Chu PB, Brener I, Pu C, Lee SS, Dadap JI, Park S, et al. Design and nonlinear servo control of MEMS mirrors and their performance in a large port-count optical switch. *J Microelectromechanical Syst* 2005;14:261–73.
- [9] Zairi S, Tian GYTY. Design of buried hinge based 3D torsional micro mirrors. *Proc Sixth IEEE CPMT Conf High Density Microsyst Des Packag Compon Fail Anal (HDP '04)* 2004:164–70.
- [10] Ataman Ç, Lani S, Noell W, Jutzi F, Bayat D, De Rooij N. A dual-axis high fill-factor micromirror array for high thermal loads. *Int Conf Opt MEMS Nanophotonics* 2011:135–6.
- [11] Michalicek MA, Clark N, Comtois JH, Schriener HK. Design and simulation of advanced surface micromachined micromirror devices for telescope adaptive optics applications. *Proc SPIE Vol 3353 Adapt Opt Syst Technol* 1998;3353:805–15.

- [12] Walton JP, Coutu R a., Starman L. Modeling and simulations of new electrostatically driven, bimorph actuator for high beam steering micromirror deflection angles 2015;9375:93750Y.
- [13] Pardo F, Cirelli R a., Ferry EJ, Lai WYC, Klemens FP, Miner JF, et al. Flexible fabrication of large pixel count piston-tip-tilt mirror arrays for fast spatial light modulators. *Microelectron Eng* 2007;84:1157–61.
- [14] Jung IW, Krishnamoorthy U, Solgaard O. High fill-factor two-axis gimbaled tip-tilt-piston micromirror array actuated by self-aligned vertical electrostatic combdrives. *J Microelectromechanical Syst* 2006;15:563–71.
- [15] Panas RM, Hopkins JB. Actuator Design for a High-Speed Large-Range Tip-Tilt-Piston Micromirror Array. *Proc ASPE 29th mtg*, Boston, MA: Nov 9-14, 2014.
- [16] Hopkins JB, Panas RM. Flexure Design for a High-Speed Large-Range Tip-Tilt-Piston Micromirror Array. *Proc ASPE 29th mtg*, Boston, MA: Nov 9-14, 2014.
- [17] Milanović V, Matus G a, McCormick DT, Ave SP, Background a. Tip-Tilt-Piston Actuators for High Fill-Factor Micromirror Arrays 2004:232–7.
- [18] Mi X, Soneda H, Okuda H, Tsuboi O, Kouma N, Mizuno Y, et al. A multi-chip directly mounted 512 MEMS mirror array module with hermetically sealed package for large optical cross-connects. *IEEE/LEOS Opt MEMS 2005 Int Conf Opt MEMS Their Appl* 2005;341:195–6.
- [19] Kim M, Park J-H, Jeon J-A, Yoo B-W, Park IH, Kim Y-K. High fill-factor micromirror array using a self-aligned vertical comb drive actuator with two rotational axes. *J Micromechanics Microengineering* 2009;19:035014.
- [20] Tsai J, Chiou S, Hsieh T, Sun C-W, Hah D, Wu MC. Two-axis MEMS scanners with radial vertical combdrive actuators—design, theoretical analysis, and fabrication. *J Opt A Pure Appl Opt* 2008;10:044006.
- [21] Dagel DJ, Cowan WD, Spahn OB, Grossetete GD, Griñe AJ, Shaw MJ, et al. Large-stroke MEMS deformable mirrors for adaptive optics. *J Microelectromechanical Syst* 2006;15:572–83.
- [22] Jia K, Samuelson SR, Xie H. High-fill-factor micromirror array with hidden bimorph actuators and TipTilt-Piston capability. *J Microelectromechanical Syst* 2011;20:573–82.
- [23] Sinclair MJ, Corporation M. 1D and 2D scanning mirrors using thermal buckle-beam actuation 2001;4592:307–14.
- [24] Jiang J, Hillerlingmann U. Design and fabrication of 2-DOF micromirror array based on electro-thermal actuators. *J Beijing Inst Technology* 2006;15:211–5.
- [25] O'Hara K. Segmented Galvanometer-Driven Deformable Mirrors. CfAO Presentations: Spring Retreat 2003. <http://cfao.ucolick.org/presentations/springretreat2003/integratedmicromachines.pdf> (accessed August 2015).
- [26] Bernstein JJ, Taylor WP, Brazzle JD, Corcoran CJ, Kirkos G, Odhner JE, et al. Electromagnetically actuated mirror arrays for use in 3-D optical switching applications. *J Microelectromechanical Syst* 2004;13:526–35.
- [27] Cho I-J, Yoon E. A low-voltage three-axis electromagnetically actuated micromirror for fine alignment among optical devices. *J Micromechanics Microengineering* 2009;19:085007.
- [28] Yee Y, Bu JU, Ha M, Choi J, Oh H, Lee S, et al. Fabrication and characterization of a PZT actuated micromirror with two-axis rotational motion for free space optics. *Tech Dig MEMS 2001 14th IEEE Int Conf Micro Electro Mech Syst (Cat No01CH37090)* 2001;00:317–20.
- [29] Standard Deformable Mirrors. Boston Micromachines. http://www.bostonmicromachines.com/production_products.htm
- [30] Cornelissen S a., Bierden P a., Bifano TG, Lam C V. 4096-element continuous face-sheet MEMS deformable mirror for high-contrast imaging. *J Micro/Nanolithography, MEMS MOEMS* 2009;8:031308.
- [31] Jung IW, Peter YA, Carr E, Wang JS, Solgaard O. Single-crystal-silicon continuous membrane deformable mirror array for adaptive optics in space-based telescopes. *IEEE J Sel Top Quantum Electron* 2007;13:162–7.
- [32] Wirth A, Cavaco J, Bruno T, Ezzo K, Road J. Deformable Mirror Technologies at AOA Xinetics n.d.;8780:1–12.
- [33] DEFORMABLE MIRRORS AND ADAPTIVE OPTIC TOOLS. Imagine Optic SA. <http://www.imagine-optic.com/en/products/deformable-mirrors-and-adaptive-optic-tools/>

Precision Machining of Yttria-stabilized Tetragonal Zirconia Polycrystal by High-speed Milling

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INTRODUCTION

Yttria-stabilized tetragonal zirconia polycrystal (Y-TZP) is a ceramic with high hardness, high wear resistance, chemical inertness, and high fracture toughness. Because of these properties and its white color, Y-TZP is a candidate for use in dental applications, such as dental crowns [1–3]. However, because of the high hardness of Y-TZP, it is rather difficult to apply a cutting process for its precision machining. At present, machining methods of Y-TZP are classified mainly into two categories. The first involves machining Y-TZP before sintering. In such a method, half-sintered Y-TZP is machined into the target shape. However, because the volume of Y-TZP shrinks during the final sintering, it is difficult to achieve the desired precise shape by this kind of method. The second category involves machining Y-TZP after sintering. In such a method, Y-TZP can be machined without considering shrinkage. However, the machining techniques are limited to grinding and polishing; the machining efficiencies of both techniques are quite low. It is also possible to cut Y-TZP by a diamond tool, but such tools are expensive.

Y-TZP has a toughening mechanism known as stress-induced transformation toughening which prevents cracks from propagating and increases the fracture toughness of Y-TZP. Inhibition of this toughening mechanism is desirable for reducing the force required for cutting Y-TZP. It is known that the toughening mechanism disappears when the temperature of Y-TZP increases above 400 °C [4] since the tetragonal structure of Y-TZP becomes stable above this temperature. Thus, the machinability of Y-TZP is expected to improve at elevated temperatures; research towards cutting Y-TZP with the assistance of an outside heat source has been reported [5–8].

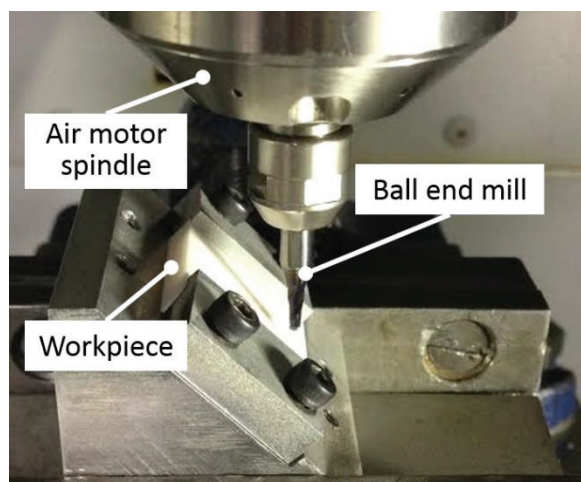
The present research aims to accomplish precision machining of Y-TZP without the use of an outside heat source or expensive cutting tools. We propose high-speed milling of Y-TZP, utilizing

the cutting heat to enhance the machinability. Our study shows that precision machining of Y-TZP is feasible.

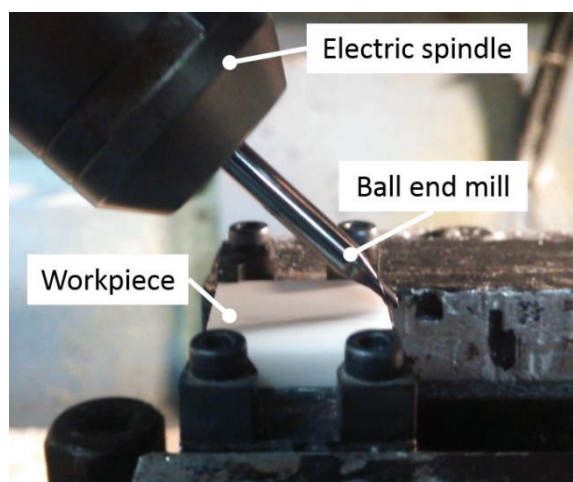
METHODS

We conducted a series of cutting experiments to investigate the effectiveness of high-speed milling. The experimental setups are shown in Figure 1(a) and (b). For high-speed milling, we used an air motor spindle (NAKANISHI Inc., HTS1501S-HSKE32) that rotates at 150,000 rpm, mounted on a three-axis machining center (DMG MORI Co., Ltd., NVD1500DCG). The experimental setup is shown in Figure 1(a). To compare the machinability using high-speed milling with that using lower-speed milling, we used an electric spindle (NSK Ltd., PSA50-135N) that rotates at speeds up to 10,000 rpm, mounted on a three-axis machining center (OKK Co., Ltd., VM4II). This setup is shown in Figure 1(b). The size of the workpiece (Nippon Tungsten Co., Ltd., NPZ-10) is 20 × 20 × 4 mm; the cutting tool (Mitsubishi Materials Co., MS2SBR0100S04) is a two-edge ball end mill having a diameter of 2 mm and made of cemented carbide with a TiAlN coating. The tool was tilted by 45° toward the feed direction in all the experiments. A schematic of the machining is shown in Figure 1(c). The workpiece and the cutting tool are shown in Figure 1(d).

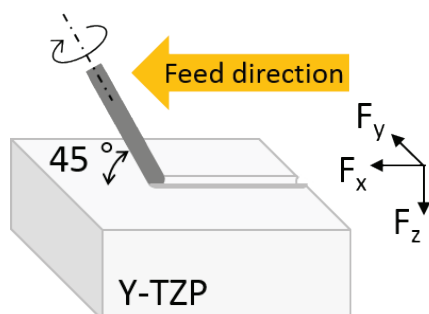
The experimental conditions are listed in Table 1. A 20-mm-long groove with a 100-μm depth was generated in every condition. Experiments 1–3 were conducted to compare the machinability of Y-TZP at cutting speeds ranging from 11 to 670 m/min. The feed rates in the three experiments were set to different values to make the feed per edge almost the same in the three experiments and to investigate the effect of the cutting speed on the machinability. Experiments 4 and 5 were conducted to investigate the effect of the feed per edge on the machinability by comparing the results with those of experiment 3.



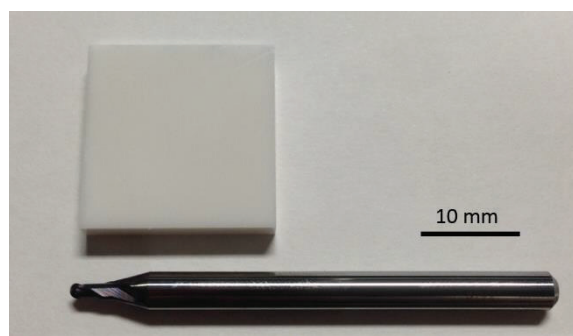
(a)



(b)



(c)



(d)

FIGURE 1. (a) Experimental setup with air motor spindle, (b) experimental setup with electric spindle, (c) schematic of the machining, (d) workpiece and cutting tool.

TABLE 1. Experimental conditions.

No.	Rotational speed, rpm	Feed rate, mm/min	Feed per edge, μm	Cutting speed, m/min
1	2500	2	0.40	11
2	10000	7	0.35	44
3	150000	100	0.33	670
4	150000	20	0.067	670
5	150000	500	1.7	670

RESULTS

Dependence on rotational speed

Figure 2 shows the top view of the grooves generated in experiments 1–3. The cutting tool was fed upward in the pictures. The surface quality improved as the rotational speed increased. Photographs of the cutting tools are shown in Figure 3. Tool wear decreased as the rotational speed increased. Figure 4 shows the dependence of the surface roughness of the generated grooves on the rotational speed.

Surface roughness was calculated from a three-dimensional image taken by a laser microscope (KEYENCE Co. VK-9510). The results show that the surface roughness improved at high rotational speeds. Figure 5 shows the dependence of the tool wear on the rotational speed. This result shows that the tool wear decreased when the rotational speed increased. These results indicate that the desired precision surface of Y-TZP can be obtained, and a long tool life can be achieved when the cutting speed is high. Figure

6 shows the measurements of cutting forces. The minimum cutting force was obtained at a rotational speed of 150,000 rpm.

Dependence on feed rate

The effect of the feed per edge on the machinability of Y-TZP at a rotational speed of

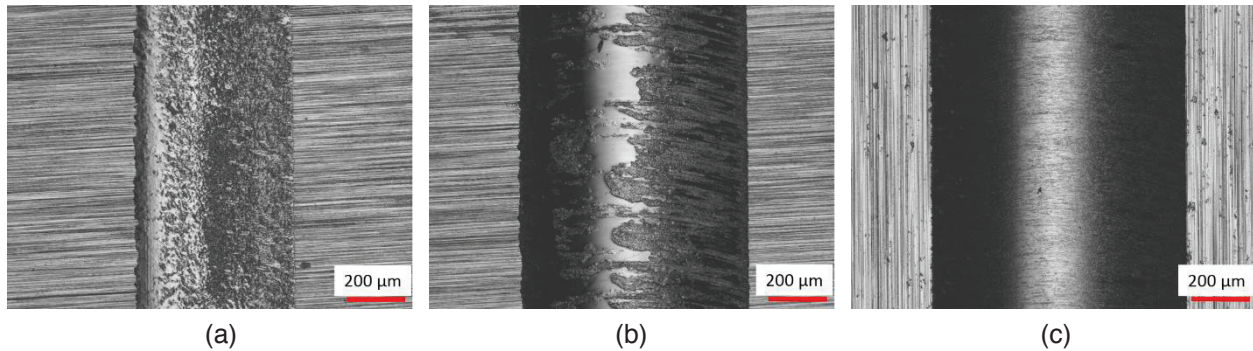


FIGURE 2. Top view of generated grooves with the rotational speed (a) 2,500 rpm, (b) 10,000 rpm, (c) 150,000 rpm.

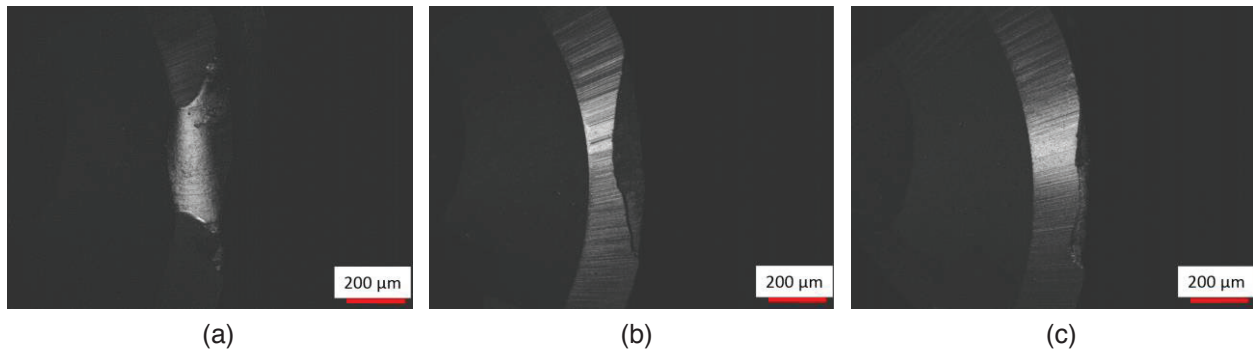


FIGURE 3. Tool wear after generating grooves with the rotational speed (a) 2,500 rpm, (b) 10,000 rpm, (c) 150,000 rpm.

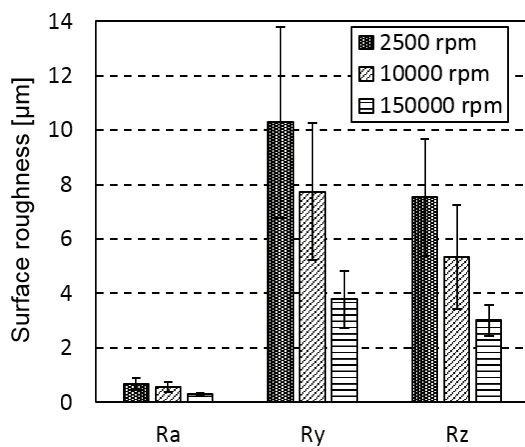


FIGURE 4. Dependence of surface roughness on rotational speed.

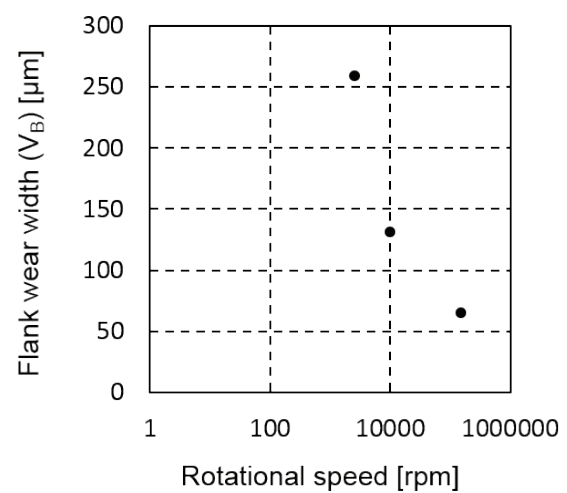


FIGURE 5. Dependence of flank wear width on rotational speed.

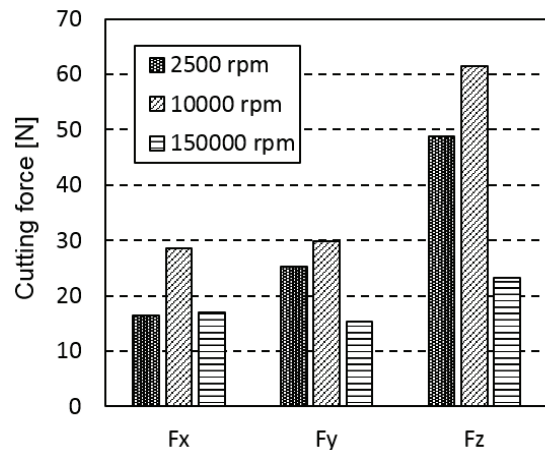


FIGURE 6. Dependence of cutting force on rotational speed.

150,000 rpm was investigated in experiments 3–5.

Figure 7 shows photographs of the bottoms of the generated grooves. Figures 7(a) and (b) show that Y-TZP was machined in the brittle mode when the feed per edge was small, while Figure 7(c) shows that it was machined in the ductile mode when the feed per edge was large. However, large cracks are evident in Figure 7(c), obtained at the highest feed rate per edge. The dependence of the surface roughness of the generated grooves on the feed rate is shown in Figure 8, which shows that machining Y-TZP in the brittle mode may yield a better surface quality than one obtained by ductile-mode cutting. Figure 9 shows the dependence of the tool wear on the feed rate. The tool life was longest when the feed rate was 100 mm/min. Results for the cutting force are shown in Figure 10. The cutting force

did not get large when the feed rate increased from 20 to 100 mm/min, but got large when the feed rate increased to 500 mm/min.

DISCUSSION

Dependence on rotational speed

The set depth of the grooves was 100 μm . Thus, the theoretical width of the grooves should be 872 μm since the diameter of the cutting tool is 2 mm. However, Figure 2 shows that the width of the grooves was 544, 784, and 875 μm , respectively, when the rotational speed was 2500, 10,000, and 150,000 rpm. This means that less volume was removed in experiments 1 and 2 than in experiment 3. We assume that this was caused by deflection and wear of the cutting tools.

The groove in Figure 2(b) was machined partially in the ductile mode, while the grooves in Figures 2(a) and (c) were machined completely in the brittle mode. The fracture toughness decreases when the temperature of Y-TZP increases. The results suggest that the temperature of the cutting zone increased and that the fracture toughness decreased when the rotational speed was 150,000 rpm, because of the heat generated by high-speed machining. This explains why the cutting force decreased drastically when the rotational speed was 150,000 rpm, as shown in Figure 6, even though the removed volume was the largest.

It is known that the hardness of yttria-stabilized zirconia decreases when the temperature increases [9]. At a rotational speed of 2500 rpm, the heat generated was not sufficient to decrease the hardness of Y-TZP, resulting in a high tool wear, as shown in Figures 3 and 5. The high tool wear resulted in a reduction in the volume of

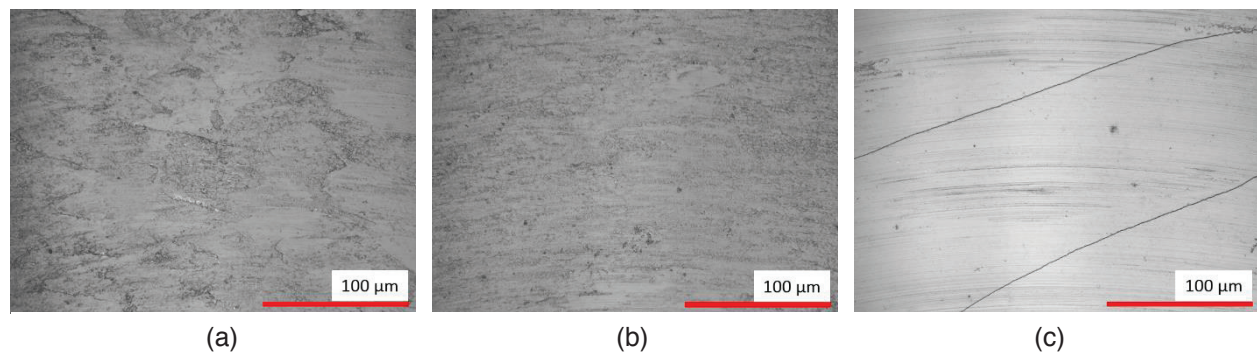


FIGURE 7. Bottom of generated grooves with the rotational speed 150,000 rpm and the feed rate (a) 20 mm/min, (b) 100 mm/min, (c) 500 mm/min.

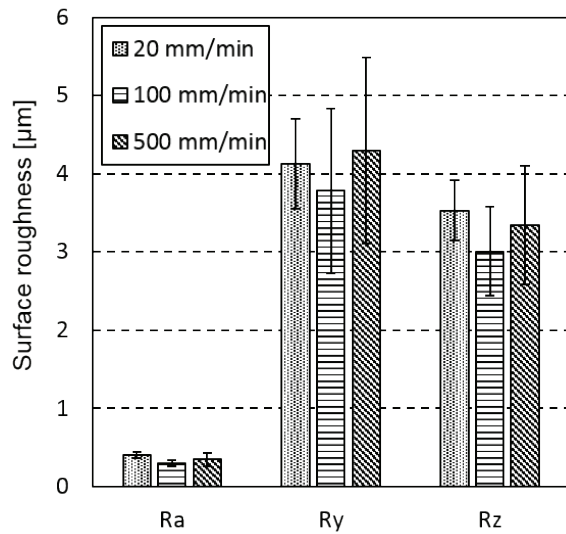


FIGURE 8. Dependence of surface roughness on feed rate.

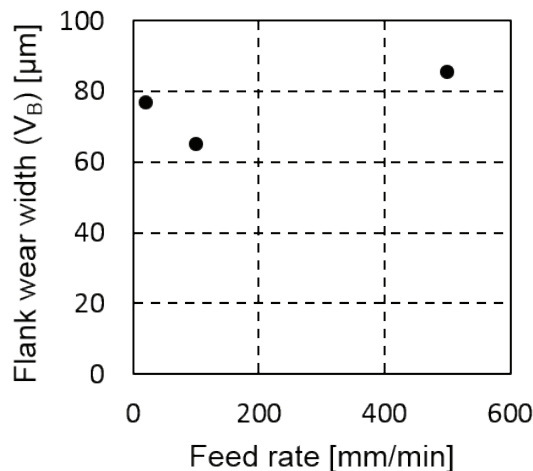


Figure 9. Dependence of flank wear width on feed rate.

material removed. Consequently, the cutting force was small, as shown in Figure 6; however, the specific cutting force was large, and thus the surface roughness was the highest at this condition, as shown in Figure 4, and the groove was machined in the brittle mode, as shown in Figure 2(a).

At a rotational speed of 10,000 rpm, sufficient heat was generated to decrease the hardness of Y-TZP, but the temperature was not high enough to cause brittle-mode machining, and thus the groove was machined partially in the ductile-

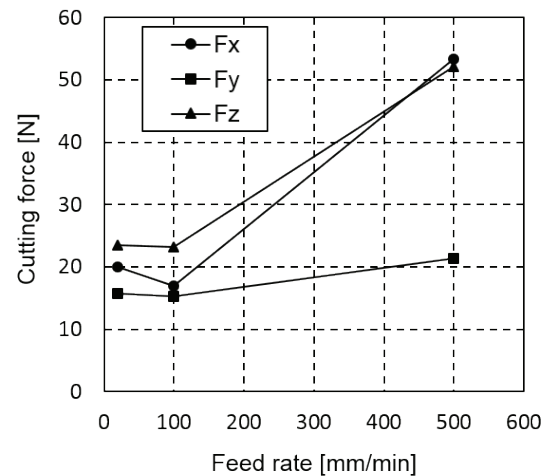


Figure 10. Dependence of cutting force on feed rate.

mode. Therefore, the tool wear decreased in comparison with the 2500-rpm condition. However, it was still much greater than the tool wear at 150,000-rpm rotational speed, as shown in Figures 3 and 5. The large tool wear caused a high surface roughness compared to results at 150,000 rpm.

At a rotational speed of 150,000 rpm, sufficient heat was generated to decrease the hardness of Y-TZP and to cause brittle-mode machining. Therefore, the cutting force and the tool wear were extremely small as shown in Figures 3, 5, and 6. Although machining occurred in the brittle mode, the surface quality and the tool life improved. This demonstrates the feasibility of precision machining of Y-TZP by high-speed milling.

Dependence on feed rate

The results above show that precision machining of Y-TZP is possible when the cutting speed is high. The machinability is affected by the feed rate as well. Figure 7 shows the bottom surfaces of the generated grooves when the rotational speed was kept constant at 150,000 rpm and the feed rate was changed. This shows that at high feed rate Y-TZP was machined in the ductile mode, which caused large cracks that are not allowable for precision cutting. This shows that the heat generated per unit volume in the cutting zone is small when the feed per edge is large.

At a feed rate of 500 mm/min, the feed per edge is 1.7 μm. The cutting force at this condition was

larger than those at feed rates of 20 and 100 mm/min, as shown in Figure 10. The large cutting force caused a significant tool wear, as shown in Figure 9, which finally degraded the surface roughness, as shown in Figure 8.

At a feed rate of 20 mm/min, the theoretical feed per edge is 67 nm. However, this is smaller than the radius of the cutting edge. Therefore, Y-TZP is machined after the edges slip on the groove several times and the depth of cut in the feed direction becomes larger than the radius of the cutting edge. This explains why the cutting force did not increase when the feed rate was increased from 20 to 100 mm/min. Normally, the cutting force increases when the feed per edge increases. We assume that the actual feed per edge feed rate at a feed rate of 20 mm/min was almost the same as that at 100 mm/min. At a feed rate of 20 mm/min, the edges slipped on the surface many times; it is assumed that the resulting friction caused excessive heat. This in turn caused increased tool wear compared with that at a feed rate of 100 mm/min, as shown in Figure 9. Figures 7(a), (b), and 8 show that the surface quality degraded compared to the quality obtained at 100 mm/min. It is assumed that the surface quality degradation was caused by the large extent of tool wear caused by the excessive heat.

At a feed rate of 100 mm/min, the tool wear was minimized and the surface quality was the best, as shown in Figures 8 and 9. Even though Y-TZP was machined in the brittle mode, a long tool life and low surface roughness were obtained.

The results in Figures 7–9 show that excessive heat degrades the tool life and surface quality. This indicates that the temperature at which brittle-mode cutting starts is the optimal temperature for precision machining of Y-TZP, and that this optimal temperature was attained under the conditions of experiment 3.

CONCLUSIONS

We propose high-speed milling of Y-TZP as a method for achieving precision machining without using an outside heat source or expensive cutting tools. Our experiments revealed that precision machining of Y-TZP was achieved when cutting occurred in the brittle mode. Specifically, the best surface quality and the smallest tool wear were obtained at a cutting speed of 670 m/min and a feed per edge of 330 nm.

ACKNOWLEDGEMENT

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REFERENCES

- [1] Guazzato M, Albakry M, Ringer S, Swain M. Strength, fracture toughness and microstructure of a selection of all-ceramic materials. Part II. Zirconia-based dental ceramics. *Dental Materials*. 2004; 20: 449-456.
- [2] Denry I, Kelly J. State of the art of zirconia for dental applications. *Dental Materials*. 2008; 24: 299-307.
- [3] Ozkurt Z, Kazazoglu E. Clinical success of zirconia in dental applications. *Journal of Prosthodontics*. 2010; 19: 64-68.
- [4] Tsubakino H, Sonoda K, Nozato R. Martensite transformation behaviour during isothermal ageing in partially stabilized zirconia with and without alumina addition. *Journal of Materials Science Letters*. 1993; 12: 196-198.
- [5] Kizaki T, Ito Y, Sugita N, Mitsuishi M. Development of laser-assisted machining of yttria-stabilized tetragonal zirconia polycrystal. *Transactions of the JSME*. 2013; 79: 4582-4592.
- [6] Ito Y, Kizaki T, Sugita N, Mitsuishi M. Laser-assisted precision machining of yttria-stabilized tetragonal zirconia polycrystals with optimal wavelength. *Proceedings of the 7th International Conference on Leading Edge Manufacturing in 21st Century*. 2013; 226-230.
- [7] Kizaki T, Ogasahara T, Sugita N, Mitsuishi M. Ultraviolet-laser-assisted precision cutting of yttria-stabilized tetragonal zirconia polycrystal. *Journal of Materials Processing Technology*. 2014; 214: 267-275.
- [8] Kizaki T, Harada K, Mitsuishi M. Efficient and precise cutting of zirconia ceramics using heated cutting tool. *CIRP Annals – Manufacturing Technology*. 2014; 63: 105-108.
- [9] Morscher G, Pirouz P, Heuer A. Temperature dependence of hardness in yttria-stabilized zirconia single crystals. *Journal of the American Ceramic Society*. 1991; 74: 491-500.

QUALITY PERFORMANCE TEST FOR SUPER PRECISION SPINDLE

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INTRODUCTION

One of the key components of a machine tool is the spindle; the machine could have a perfect structure and guide-ways, but if the spindle axis moves, the work-piece will present geometrical form errors [1]. Through the years several methods have been developed to characterize spindle axis motion from crude run out tests [2-3], to several contact and non-contact methods using LVDT's, inductive probes and capacitance probes [5]. Today the most accepted method uses a master target and capacitance probes instrumented and connected to a personal computer where all the calculations are made. This modern axis motion measuring instrumentation is capable of sub-micron resolution and accuracy. This equipment mostly finds its place in air bearing and hydrostatic spindle research and manufacturing, since the machines where these are used require ultra-precision components. While this may be true, modern spindle axis motion measuring is also a valuable resource for the Super Precision® machine tool manufacturers[4]. The information acquired from these measurements can help in finding spindle defects and their sources. It can also identify if certain assembly process changes have a positive or negative impact. In the following paragraphs it will be shown how the measurement of spindle axis motion provides valuable advantages to the Super Precision® spindle manufacturer.

PURPOSE OF MEASURING SPINDLE AXIS MOTION

The part roundness is determined by a test cut performed on the completed machine. At this point the spindle has been assembled on the machine, where several covers and support systems have been installed. If the spindle does not meet the required specification, it will be rejected and several hours of assembly time are wasted. This creates the need for a spindle quality performance test. For a Super Precision® turning center the roundness should be under $0.50\mu\text{m}$ ($20\mu\text{in}$). The manufacturing process to achieve this roundness has been established and for the most part is stable.

Nevertheless if a nonconforming spindle is detected before it can be installed on the machine, unnecessary waste is avoided. Measuring the roundness of the spindle requires a source of driving power, a work-holding device for the sample part and a guiding system for the tool. Supplying all these elements for a spindle which is not installed on the machine tool is impractical, cumbersome and can introduce additional errors to the roundness measurement. Taking these factors into consideration, it was determined that the most feasible characteristic to measure when the spindle is outside the machine tool is the spindle axis motion; which subsequently can be correlated to spindle roundness.

The spindles being measured are used in a super precision turning center. The spindle is driven by a multi-strand V belt transmission when installed on the machine. During the production process the spindle requires a run-in procedure to purge the grease. At this stage performing a spindle measurement is ideal since it can be easily instrumented and driven. The main objective, after the spindle axis motion has been acquired, is to find the relationship between this information and the part roundness.

HARDWARE AND MEASUREMENT

Two key parameters that need to be measured are: part roundness, which is the output, and spindle error axis motion, which is the input. The part roundness measurement is simple and straight forward. A C360 brass 1 inch diameter specimen is cut at 1000rpm using a natural diamond at a feed rate of .0009ipr. The length of the cut is also 1 inch and 3 roundness measurements are taken: top middle and bottom. The average of these three measurements is the part roundness value used and recorded in the inspection files, see figure 1.



Roundness measurements

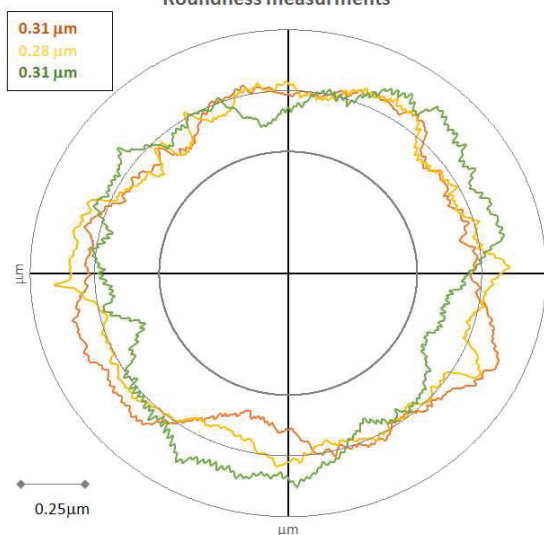


FIGURE 1. Roundness measuring machine Rondcom 54 (top). Output of roundness measurement (bottom), the three measurements are used to compute the cut average roundness.

The spindle axis motion is measured using a 3 channel LION precision© spindle analyzer when mounted on the run-in bench. In general terms this equipment is a set of capacitance probes that measure the relative distance from a master target when the spindle is rotating. All this data is processed by the spindle analyzer software [5]. The capability and amount of information that can be obtained from this device are immense, however, we will concentrate on the radial motions of the spindle since they are related to part roundness.

The capacitive sensors are secured by a probe nest, which is mounted to a fixture that has fine adjustment screws for alignment; as shown in figure 2.

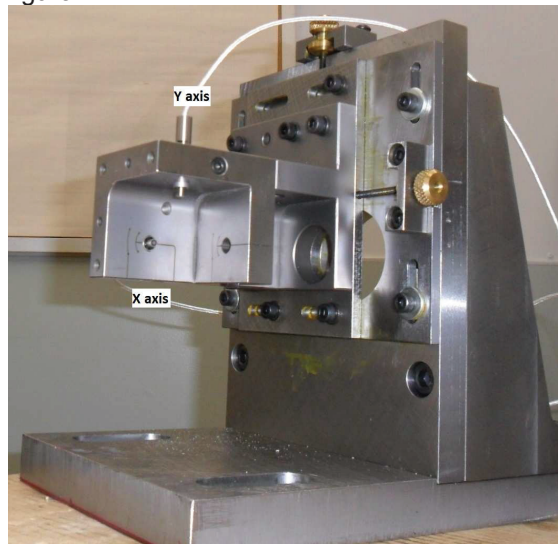


FIGURE 2 Run-in bench fixture used to hold the probe nest. Note the oversized steel plates used.

To obtain the bench natural frequency a bump test is performed. When the fixture is secured to the running bench the natural frequency of the setup is 64 Hz which is higher than the testing rpm frequency of 16Hz. This is key to avoid any resonance of the fixture structure that could potentially affect the measurements [6].

In addition, the noise of six different spindle setups in the running bench with the spindle stopped was measured. It was found that the average peak to peak value of the noise is $0.07\mu\text{m}$. Since this value is an order of magnitude smaller than what we are looking for, the measurements can be considered reliable. The noise contains some frequencies on the 60 Hz range and its harmonics, most likely introduced by the surrounding electrical systems. In case the idle noise levels are on unacceptable range, the measurement setup is checked and the detected noise sources are eliminated.

The signals from the X & Y sensors are logged at a sampling frequency of 5kHz. The data of these two channels are plotted on a rotating sensitive direction graph by the spindle analyzer software. This graph is visually inspected to detect any major anomalies or problems. Our main interest is the data collected by the X axis

probe, which is the sensitive direction. For a lathe the sensitive direction is where the tool is perpendicular to the axis of rotation on the X-Y plane. Any spindle motion along this axis will have a direct effect on machined part roundness [1].

Several approaches exist to analyze the error axis motion, from simple time domain techniques to advanced frequency domain filtering [1],[6]. The synchronous (or average) error motion has been selected as the main analysis parameter, due to its stability and self filtering nature. The synchronous error is the component of the total error motion that occurs at integer multiples of the rotation frequency [5]. It is typically obtained by averaging a certain number of revolutions and calculating the distance between the minimum and maximum inscribed circles. From the physical point of view it also makes sense to use the synchronous error, because during the cutting process most of the asynchronous errors will be part of the surface finish[1] and not the form.

To simplify the axis motion measurement, no encoder is used; the angle of rotation is obtained by leaving a small amount of eccentricity (10 μ m) on the master target. This adds a sinusoidal component to the sensor data, which is removed by fitting a least squares sine wave to the data. The fitted sine wave is also used to obtain the rotation angle of the spindle.

The number of turns used to calculate the synchronous error motion is not a trivial matter. The behavior between the synchronous error magnitude and the number of turns used is exponential, as shown in figure 3. Taking this into consideration, if not enough turns are used, the synchronous error will be too high. On the other hand if too many turns are used information from the synchronous error will be lost due to the averaging. A good balance was found when using 10 turns to calculate the synchronous error motion, since its where the exponential starts to flatten out. In addition the default value for number of turns used on calculations on the LION© spindle analyzer is also 10.

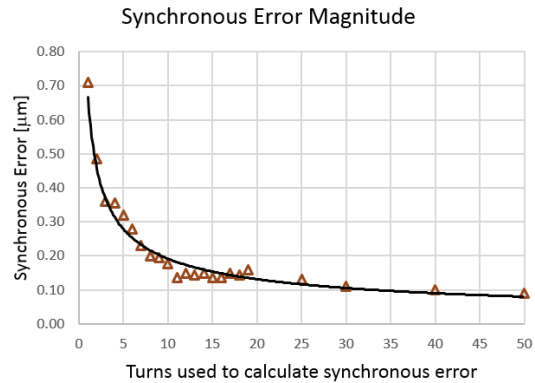


FIGURE 3. Using the same axis motion data set the synchronous error was calculated for different number of turns, an exponential behavior is readily observed.

EXPECTED ROUNDNESS MODEL

Our main objective is to relate the measured axis motion of the spindle at the running bench with the machine roundness cut average. The output of the machine roundness cut depends on several factors, some are directly related to the spindle itself and others depend on the machine assembly. This is important to understand because there is a source of variation which is not related with the measured spindle axis motion. To minimize the effect of this variation the roundness cuts are taken after the machine has been properly balanced and aligned. A spindle is considered unacceptable if the average roundness cut is above 0.50 μ m.

To obtain a value for the expected roundness the spindle axis synchronous error was calculated by dividing each revolution in 2 degree intervals, averaging the motion for 10 turns, and taking the difference between the maximum and minimum averages. This calculation is considered as the mean roundness; however, this value is not a full representation of the process. With this in mind, the values that define the maximum and minimum over the 10 turns are used to obtain the standard deviation. With the means and the standard deviations the Student t-distribution [7] is used to calculate the upper interval of confidence for the maximum and the lower for the minimum. The difference between these intervals is our expected roundness. Figure 4 shows the steps used to calculate the expected roundness.

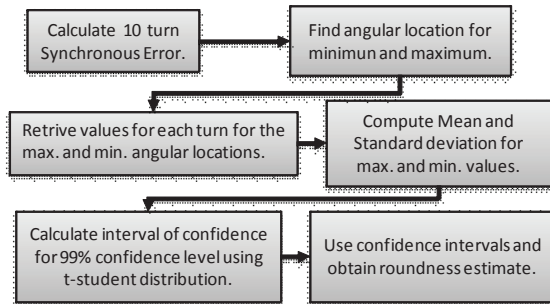


FIGURE 4. Algorithm used to obtain roundness estimate with a 99% confidence level.

A confidence level of 99% was selected since the main objective is to avoid installing a bad spindle on a machine. When this happens a great amount of time and money are wasted.

TEST RESULTS

The proposed approach was applied on 7 production spindles, by measuring the axis motion on the running bench computing the expected roundness and then measuring the roundness from test cuts. These results are shown in figure 5.

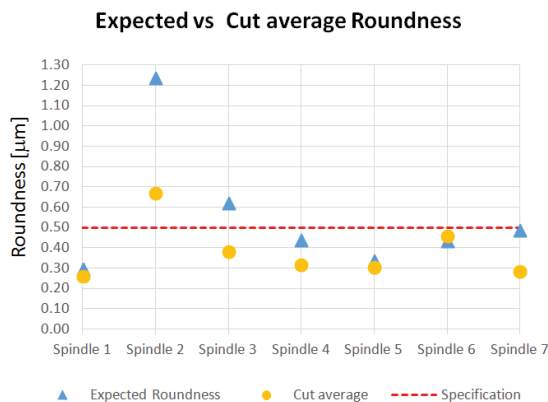


FIGURE 5. Expected roundness and actual cut part roundness obtained from the average of three measurements.

It can be observed that six of the seven expected values were accurate in predicting if the spindle would pass or fail the roundness cut. Spindle 3, where the expected roundness indicated that the spindle would not pass, shows that the method works as a failsafe. Since qualifying a bad spindle as good is much more costly than the opposite scenario. When the prediction is a higher roundness value than the maximum allowed ($0.50\mu\text{m}$), the spindle is further checked to determine if a fault exists.

CONCLUSIONS

When using the spindle error axis motion an expected roundness can be computed with a confidence level of 99%. This provides a valuable tool for manufacturing, avoiding the assembly of spindles that will not pass the roundness cut. It is understood that some inherent variability comes from the machine assembly and not only from the spindle axis motion. Examples of machine assembly factors that affect roundness include: balancing, axis vibration, machine stiffness, part material, tool material, cutting parameters, auxiliary equipment vibration, etc. Taking this into consideration another benefit of the proposed expected roundness computation is that it can help to pin point if the problem is on the machine or spindle assembly.

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REFERENCES

- [1] J.B. Bryan, P. Vanherck, "Unification of Terminology concerning the error motion of axes of rotation" June 20, 1975, CIRP
- [2] E. J. Goddard, A. Cowley, M. Burdekin, "A measuring system for the evaluation of spindle rotation accuracy" September 18, 1972 13th International machine tool design and research conference.
- [3] G. Schlesinger, "Testing of machine Tools", 1966 7th ed, Machinery Publishing Co. Ltd.
- [4] E. Kushnir, "Super-precision turning center: high accuracy, hard turning, multi-tool machine tool" October 20, 2013 ASPE
- [5] Lion Precision, "Instruction Manual Advance Spindle Error Analyzer v7", 2003
- [6] E. R. March, "Precision Spindle Metrology" 2010, 2nd ed, DEStech Publications
- [7] D. C. Montgomery, "Introduction to Statistical Quality control", 2009 6th ed, John Wiley & Sons

ANALYTICAL MODEL FOR THIN PLATE DYNAMICS

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INTRODUCTION

In many industries, such as aerospace, high-speed milling of flexible components is a common manufacturing process. Productivity in milling thin, monolithic components is often limited by regenerative chatter [1-2]. The present study concentrates on the thin plate dynamics used to model the flexible components. An analytical model is developed for the minimum lateral stiffness of thin ribs with clamped-clamped-clamped-free (CCCF) boundary conditions. The dynamic characteristics of the plate were predicted using finite element analysis and its accuracy was experimentally validated.

The dynamic characteristics of these structures are conventionally described using the complex-valued frequency response function, or FRF, which defines the vibration output to force input ratio in the frequency domain. It represents the steady-state solution to the system differential equation of motion [3, 4].

In this paper, the CCCF plate structure is modeled using finite element software Abaqus/Standard 6.13. The most flexible mode and location in the plate structure are considered when predicting the corresponding stiffness and natural frequency. The analytical model is intended to assist in peripheral milling of flexible structures. The paper is organized as follows. Modeling and identification of the FRF and mode shapes for plates with cantilever (case 1) and CCCF (case 2) boundary conditions using the finite element method, or FEM, are first presented. Next, the experimental procedure is described. Finally, a comparison between predicted and experimental results is presented.

FINITE ELEMENT MODELING

Numerical analysis may be used rather than analytical techniques for complex structures. FEM is the most widely used numerical technique for structural analyses. In FEM the continuum or

domain is divided into a series of finite number of regions, called finite elements, which neither overlap nor have a gap between each other. The behavior of each element is controlled by the number of key points on each element called nodes. The displacement or stresses at any point in an element be assigned to these nodes which enables a problem with an infinite number of degrees of freedom to be converted to one with a finite number using differential equations of motion for the nodes. The motion of a dynamic structure or system may be represented by a set of simultaneous differential (coupled) equations using FEM. These coupled equations of motion are solved by transforming them into a set of independent (uncoupled) equations by means of modal matrix. This procedure is called numerical modal analysis [5]. Modal analysis is a process for determining the system's modal parameters, including natural frequency and mode shape. Finite element analysis offers an effective predictive capability for modelling the thin-walled structures.

Analysis procedure

In Abaqus/Standard, modal analysis is referred to as dynamic analysis in which the response of linear systems is calculated on the basis of the necessary modes and frequencies. The procedure for dynamic analysis is summarized as follows.

Natural frequency extraction

Eigenvalue extraction is used to calculate the natural frequencies and the corresponding mode shapes of a system using the Lanczos eigensolver. The eigenvalue problem for natural frequencies of an undamped finite element model is:

$$(-\omega^2 M^{MN} + K^{MN}) \phi^N = 0, \quad (1)$$

where M^{MN} is the symmetric and positive definite mass matrix; K^{MN} is the stiffness matrix; ϕ^N is the eigenvector; and M and N are the degrees of freedom [6].

A first step for natural frequency extraction in this study was to build a model with the desired geometric dimensions (Fig. 1) and then assign the material properties and boundary conditions (see Fig. 2). Next, for meshing the model, an eight-node, three dimensional elements with reduced integration (C3D8R) was used. Finer mesh was used for the thin-walled structure (rib) while a coarser mesh was used in the other areas (base). A total of 91908 and 126456 elements were generated for the cantilever and CCCF boundary condition models.

The predicted mode shapes and corresponding natural frequencies for the cantilever and CCCF boundary conditions are shown in Figs. 3-7.

Mode-based steady-state dynamic analysis

After extracting the natural frequencies and mode shapes, the mode-based steady-state dynamic analysis was used to calculate the steady-state dynamic linearized response of each system, i.e., the FRF for harmonic excitation. As the name suggests, this step calculates the response based on the system's eigenfrequencies and modes and, therefore, requires that a natural frequency extraction procedure be performed prior to the steady-state dynamic analysis. The mode-based steady-state dynamic step is defined by specifying the frequency ranges of interest with a linear frequency spacing and a bias parameter of 1. Also, the damping coefficient, i.e., the viscous damping ratio, is defined for a specified mode number and a concentrated nodal force is applied to the displacement degree of freedom at the location of most interest in the structure's response. These loads vary sinusoidally with time over a user-specified range of frequencies [6].

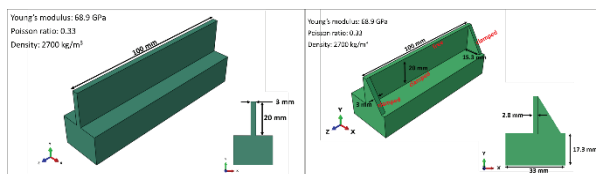


FIGURE 1. A schematic of the monolithic structure with cantilever (left) and CCCF (right) boundary conditions. The inset shows a cross-section of the system and the material properties for 6061-T6 aluminum.

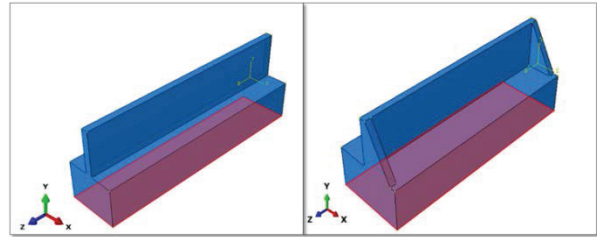


FIGURE 2. FEM encastre boundary conditions for cantilever (left) and clamped-clamped-clamped-free (right) boundary conditions: the bottom face is constrained from all displacements and rotations.

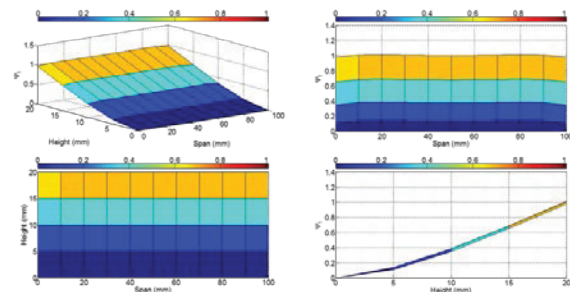


FIGURE 3. Predicted first mode shape with a natural frequency of 5605.2 Hz for cantilever boundary condition.

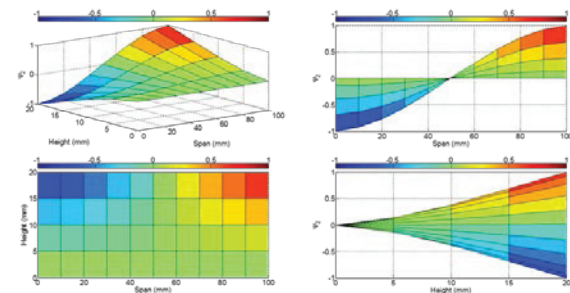


FIGURE 4. Predicted second mode shape with a natural frequency of 6115 Hz for cantilever boundary condition.

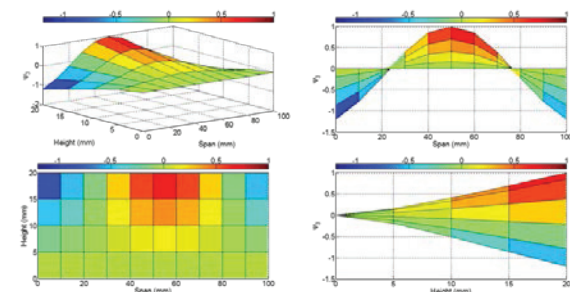


FIGURE 5. Predicted third mode shape with a natural frequency of 7668.5 Hz for cantilever boundary condition.

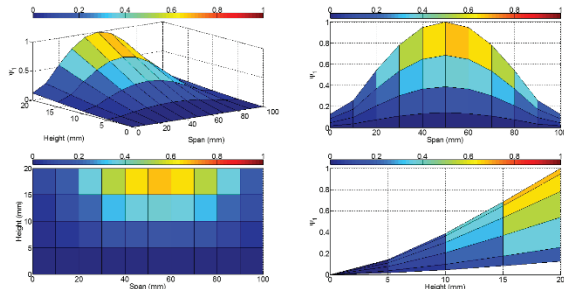


FIGURE 6. Predicted first mode shape with a natural frequency of 5821.9 Hz for CCCF boundary condition.

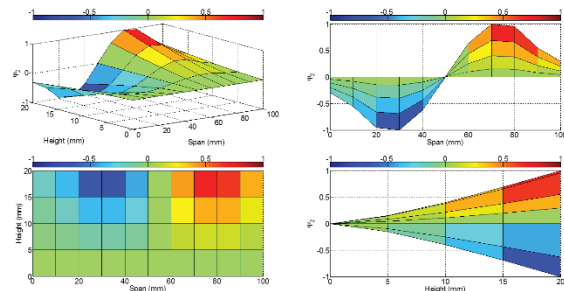


FIGURE 7. Predicted second mode shape with a natural frequency of 7598.1 Hz for CCCF boundary condition.

EXPERIMENTAL SETUP

The plate FRFs were measured using the commercial software package MetaMax. The input force was applied using a modal hammer (PCB 084A17) with a steel tip (PCB 086E80 SN 33416) and the vibration response was measured with a laser vibrometer (Polytec OFV-534); see Figs. 8-9. The vibrometer controller was set for a measurement range of 50 mm/s/V and a 1.5 MHz upper frequency.

In order to obtain FRFs that accurately reflect the predicted workpiece dynamics, it is important to very closely match the workpiece's boundary conditions. Therefore, the workpiece was glued using cyanoacrylate adhesive to a heavy steel table. The laser vibrometer target spot was located at the top right corner of the plate under test with approximately a 295 mm stand-off distance and the structure was impacted with the hammer at multiple locations. Given the time delay between the laser vibrometer and hammer signals, a phase correction algorithm [4] was used to remove the corresponding phase error from the measured FRF.

FEA AND EXPERIMENTAL RESULTS

The predicted and measured mode shapes, stiffness along the free edge, and FRF for cases 1 (cantilever) and 2 (CCCF) are shown in Figs. 10-17.

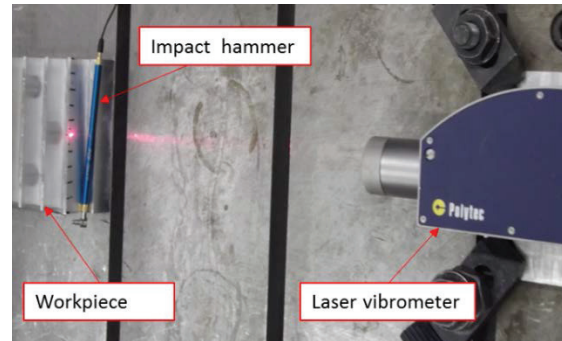


FIGURE 8. Plate with cantilever boundary condition FRF measurement via impact hammer and laser vibrometer.

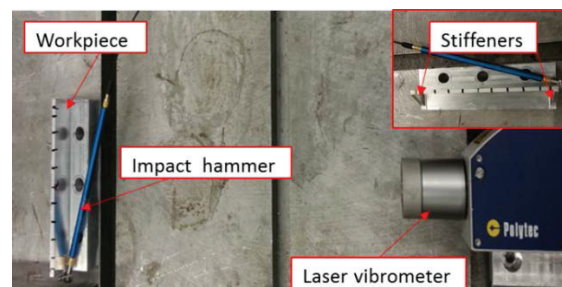


FIGURE 9. Plate with CCCF boundary condition FRF measurement via impact hammer and laser vibrometer. The inset shows stiffeners which provide clamped boundary condition at each end.

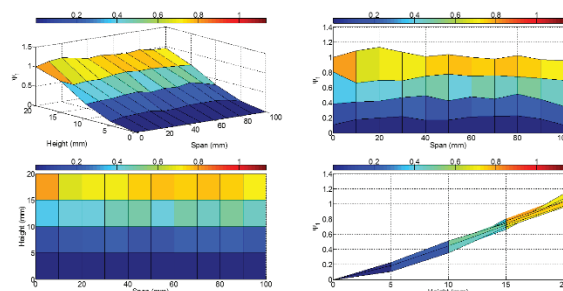


FIGURE 10. Measured first mode shape with a natural frequency of 5774.7 Hz for cantilever boundary condition.

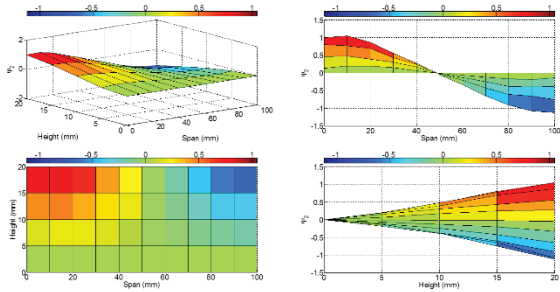


FIGURE 11. Measured second mode shape with a natural frequency of 6319.4 Hz for cantilever boundary condition.

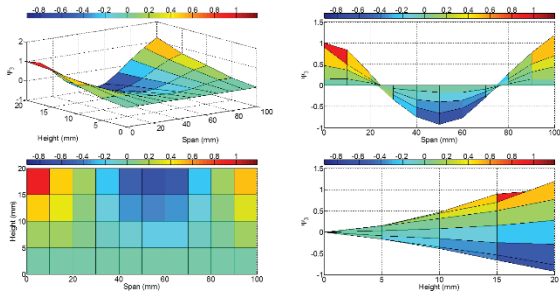


FIGURE 12. Measured third mode shape with a natural frequency of 7939.1 Hz for cantilever boundary condition.

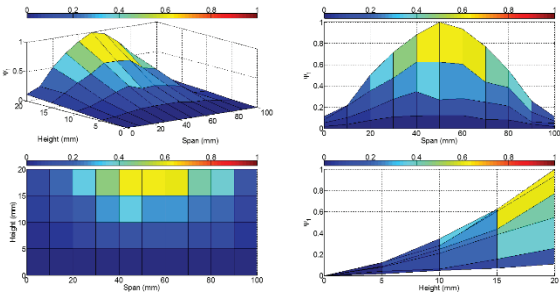


FIGURE 13. Measured first mode shape with a natural frequency of 6002 Hz for CCCF boundary condition.

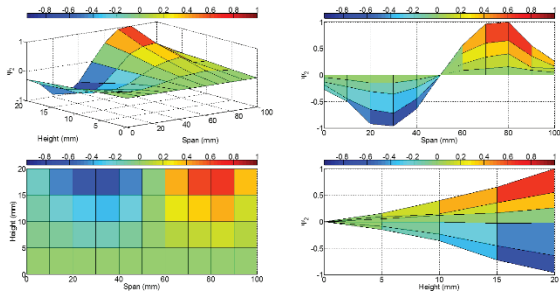


FIGURE 14. Measured second mode shape with a natural frequency of 7795.7 Hz for CCCF boundary condition.

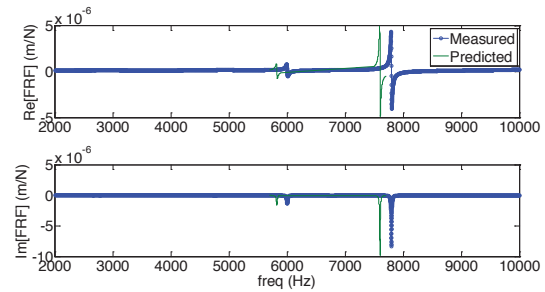


FIGURE 15. Comparison of the predicted and measured FRF for CCCF boundary condition.

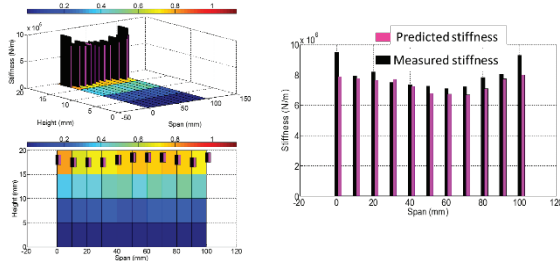


FIGURE 16. Comparison of the predicted and measured first mode stiffness at the free edge for the plate with cantilever boundary condition.

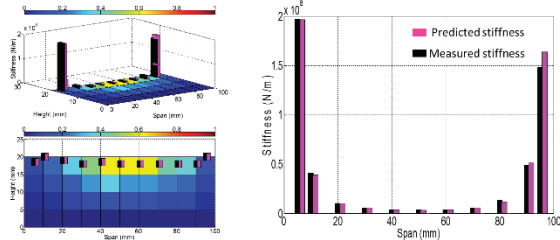


FIGURE 17. Comparison of the predicted and measured first mode stiffness at the free edge for the plate with CCCF boundary condition.

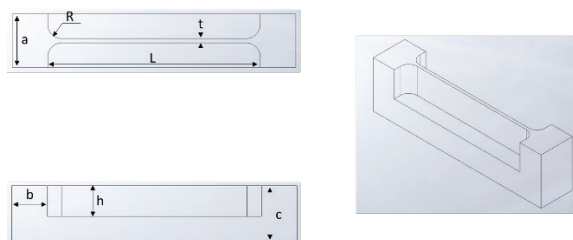


FIGURE 18. Detailed drawing for different test geometries.

TABLE 1. Geometry parameters and material for different test geometries (all dimensions in mm).

Case Geometry	3	4	5
A	25	25	37.72
B	14.0	13.7	24.76

c	36.91	47.86	53.75
R	11.43	11.43	10
L	150	150	150
t	2.01	2.01	2.95
h	30	40	30
Material	Al6061	Al6061	Ti6Al4V

Further analyses were conducted on other plate geometries with CCCF boundary conditions. Natural frequencies are shown in Fig. 19 and Table 1.

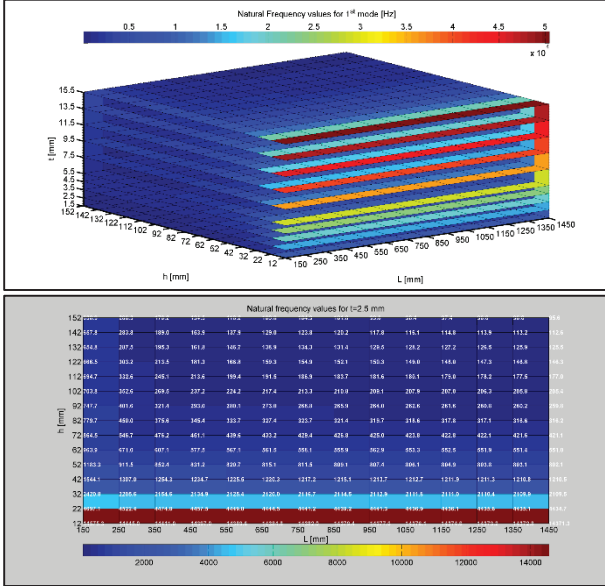


FIGURE 19. Natural frequency values for (top) entire range (bottom) 2.5 mm thickness only.

TABLE 2. Comparing the first mode predicted and measured modal parameters for different test geometries.

Test geometry	Case 1	Case 2	Case 3	Case 4	Case 5
Modal Parameters					
Measured natural frequency, f_{n_m} (Hz)	5757.9	5994.4	2238.5	1468.7	3123.5
Predicted natural frequency, f_{n_p} (Hz)	5605.7	5822.1	2173.4	1415.2	3100.3
$(f_{n_m} - f_{n_p})/f_{n_m} \times 100$ (%)	2.7	2.9	2.9	3.6	0.74
Measured modal stiffness, k_m (N/m) ($\times 10^6$)	7.2581	3.721	0.5311	0.302	2.6382
Predicted modal stiffness, k_p (N/m) ($\times 10^6$)	6.7672	3.406	0.5514	0.2904	2.5451
$(k_m - k_p)/k_m \times 100$ (%)	6.8	8.5	-3.8	3.8	3.5

CONCLUSIONS

In this study, a comparison of predicted and measured natural frequencies, stiffness values, and mode shapes showed good agreement for thin-walled structures with cantilever and clamped-clamped-clamped-free boundary conditions. Using the numerical finite element modeling capability,

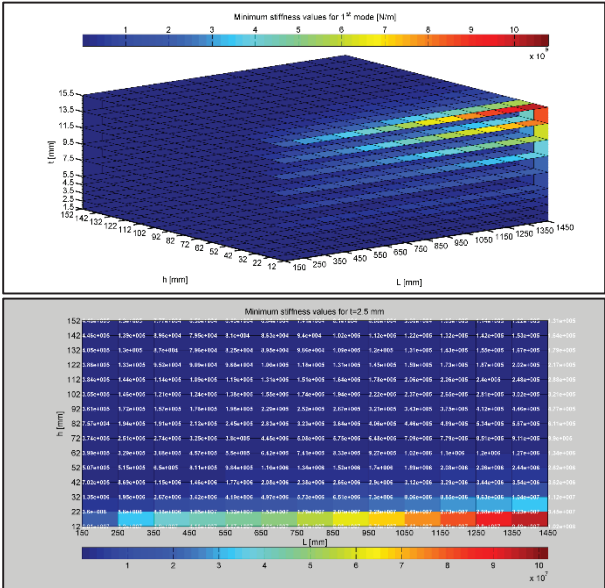


FIGURE 20. Minimum stiffness values for (top) entire range (bottom) 2.5 mm thickness

Finally, using the numerical FEA modeling capability, the minimum stiffness and its corresponding natural frequency (for first mode) was evaluated for changes in length ranging from 150 to 1450 mm, thickness from 1.5 to 15.5 mm, and height from 12 to 152 mm. The result is shown in Fig. 20. All results are summarized in Table 2.

stiffness values, the workpiece dynamics can be defined and, using milling stability analysis methods, stable machining parameters may be identified.

REFERENCES

- [1] Sébastien Seguy, Gilles Dessein, Lionel Arnaud, Surface roughness variation of thin wall milling, related to modal interactions, *International Journal of Machine Tools and Manufacture* 48 (2008) 261-274.
- [2] Scott Smith, Robert Wilhelm, Brian Dutterer, Harish Cherukuri, Gaurav Goel, Sacrificial structure preforms for thin part machining, *Annals CIRP* 61 (2012) 379-382.
- [3] T. Schmitz, S. Smith, *Mechanical Vibrations: Modelling and Measurements*, Springer, New York, NY, 2012.
- [4] V. Ganguly, T. Schmitz, Phase correction for frequency response function measurements, *Precision Engineering* 38 (2014) 409-413.
- [5] Zu-Qing Qu, *Model Order Reduction Techniques with Applications in Finite Element Analysis*, Springer Science and Business Media, 2013.
- [6] Dassault Systemes Simulation Corp., *User's Manual Version 6.13-2*, Providence, RI, 2013.

Multi-DOF Manipulator with Force Feedback for Target Assembly of HEDS Targets

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OVERVIEW

We have developed a multiple degree-of-freedom manipulator intended for high precision assembly of millimeter scale parts. Precision alignment and assembly is critical for the proper fabrication of High Energy Density Science (HEDS) targets whose alignment requirements are on the scale of single microns [1] [2]. In addition, components are often fragile, limiting the amount of force that can be applied during assembly. Increased precision and control facilitates the fabrication of improved targets, resulting in higher quality and more consistent data. Target fabrication is an active area of development for HED research [1] [2].

The manipulator has 6 degrees of freedom with a 12.5mm range in the 5 translation axes and a full 360 degree rotation about the Z axis. The measured z axis resolution is 100nm while the force/torque sensor has a 3.9mN and 50mN-mm resolution. Power spectral density analysis of the system indicates two noise sources with sensor drift becoming significant in operations over 2hrs.

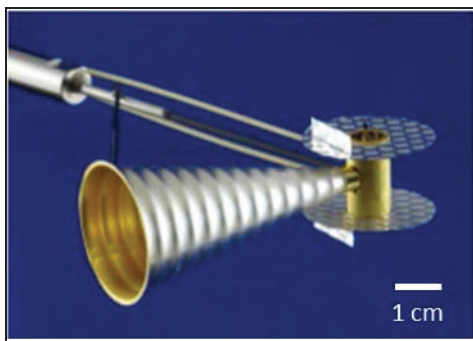


FIGURE 1: Example of target fabricated for high energy density equation of state testing.

MANIPULATOR OVERVIEW

The manipulator is comprised of two 3 degree-of-freedom stages mounted in a vertical orientation such that parts being assembled can

be manipulated to create a collinear axis between two reference surfaces. The use of two 3-axis stages is driven by the need for a fine controller (top) and a course, large range stage for part transfer in and out of the operating volume (bottom).

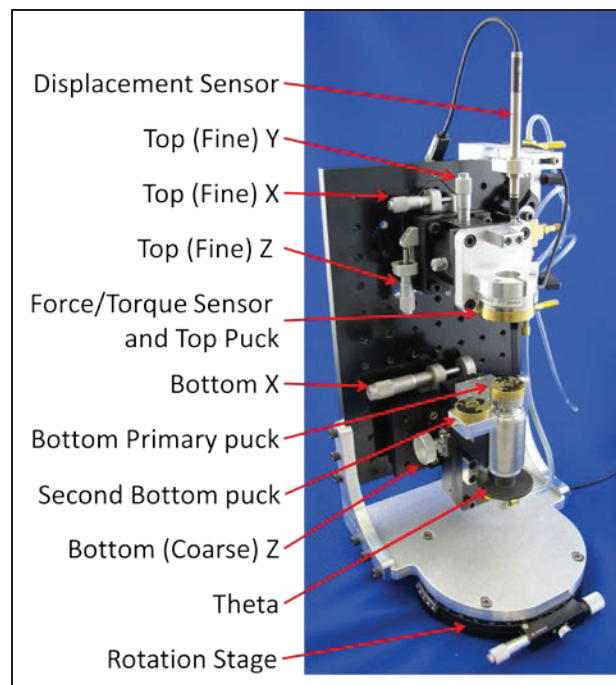


FIGURE 2: Multi-DOF Manipulator with Force Feedback. Two stacked stages are used to manipulate parts in relation to each other for precision assembly. A multi-axis force/torque sensor and Z displacement sensor provide feedback to the user.

The top stage system has 3 axes controlled via micrometers with movement ranges of 12.5 mm in X, Y and Z directions. A six axis force torque sensor is mounted to the top set of stages. It has an 18N force range (X,Y,Z) with a 3.9mN resolution and a 250 N-mm torque range (tZ,tY,tX) with a 50 mN-mm resolution. A single axis, z, displacement sensor is pressing against the top stage to measure how far it moves

during assembly. The displacement sensor has a 12mm range with a $0.1\ \mu\text{m}$ resolution and $1\ \mu\text{m}$ accuracy. These two sensors allow the user to monitor and record multi-axis loads and Z displacements during the assembly process.

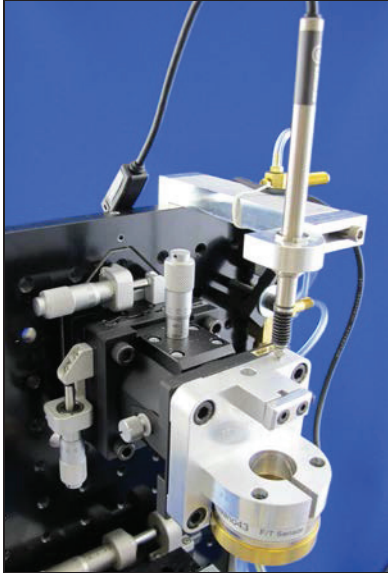


FIGURE 3: Top stage system. Micrometers control stages in X,Y and Z with 12.5 mm ranges. A six axis force/torque sensor is mounted to the stage system. A brass puck holds parts to the F/T sensor and stage. A displacement sensor measures the Z movement of the stage system.

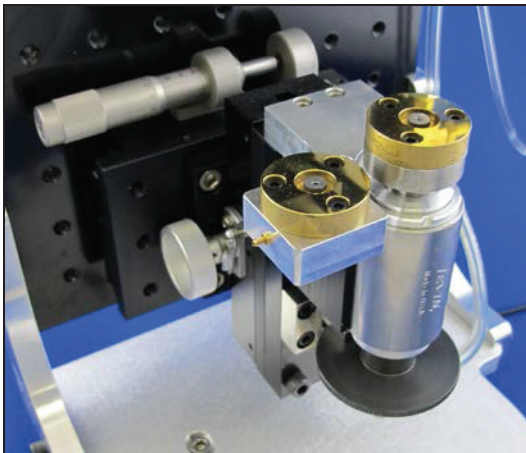


FIGURE 4: Bottom stage system. A course Z stage with a 50mm range allows for part loading on both top and bottom pucks. An X stage controlled by micrometer allows for a 2nd puck to be used in multi-component assemblies. The primary bottom puck is capable of 360 degree rotation about the Z axis.

The bottom positioner provides both θ_z motion of the part (360 degrees) as well as a large range coarse Z displacement (50.0 mm). The large Z range is intended to provide sufficient space for manual placement of parts on the stages' end effectors. The bottom stage pictured in Figure 3 contains a second puck and the ability to move in the X direction such that either puck can be aligned below the top stage (12.5 mm). This facilitates the multi-step assembly process.

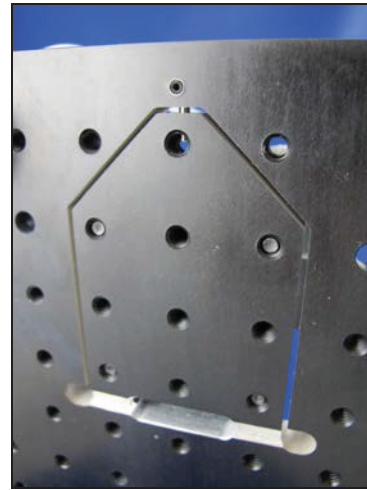


FIGURE 5: Alignment Flexure. The top stage is mounted to this single degree of freedom flexure. The flexure is used to adjust the θ_x orientation of the top stage relative to the bottom stage.

There is no on-the-fly θ_x or θ_y rotation capability. Each manipulator must be adjusted upon fabrication to align the central axis of the top stage set to the central axis of the bottom stage set. A single axis flexure is cut into the back plate to which the top stage is attached. The θ_x angle of the stage is adjusted through the set screw pressing on the pin attached to the flexure. θ_y is adjusted using the slop in the screws mounting the top stage to the back plate.

SEMI-KINEMATIC MOUNTING SYSTEM

Repeatable alignment of parts is critical to consistent assembly. In the LLNL HED target fabrication group, almost every machine is equipped with a semi-kinematic puck/pedestal system. The puck is a diamond turned brass disk with a precision ground tungsten carbide half-sphere mounted in the center. These pucks are mounted and dialed into all assembly stages

and machinery. Pedestals are aluminum holders to which components and materials are mounted. The pedestals create a safe handling area for fragile components and materials. They also repeatably mount to the machinery mounted pucks allowing parts to be transferred between processes. Brass and aluminum were chosen for the puck and pedestal because they are dissimilar metals, thereby reducing the chance of galling when repeatedly swapped.

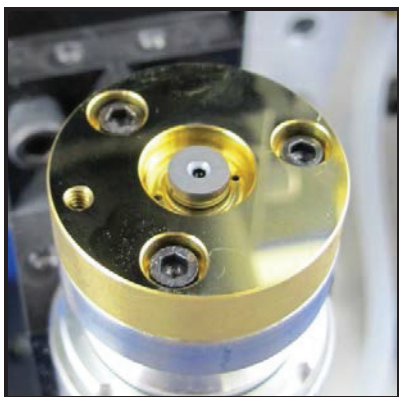


FIGURE 6: Brass puck mounted to the manipulator bottom stage system. In the center is the tungsten-carbide half-sphere.

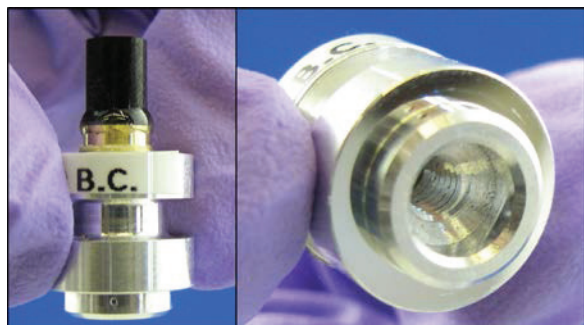


FIGURE 7: Aluminum pedestal with CRF mounted to the top. The bottom surface of the bottom ring and the inside of the cylinder are diamond turned to mate with components of the puck.

The puck/pedestal mating creates a semi-kinematic system to allow repeatable movement of parts between machining operations and assembly operations. The diamond turned cylindrical surface (Figure 8, item 2) on the pedestal mounts around the tungsten carbide sphere on the puck, constraining 3 degrees of motion. Two more degrees are constrained by the mating of the diamond turned flats on both pieces (Figure 8, item 1). The final degree of

freedom (θ_z) is not constrained allowing for the parts to be rotated on machines that don't have an axis of rotation. A mark on the pedestal and on the puck allows for repeatable alignment in the θ_z direction if desired. The semi-kinematic coupling is repeatable to $2\ \mu\text{m}$ with the same puck/pedestal combination and $10\ \mu\text{m}$ between a single pedestal and several pucks.

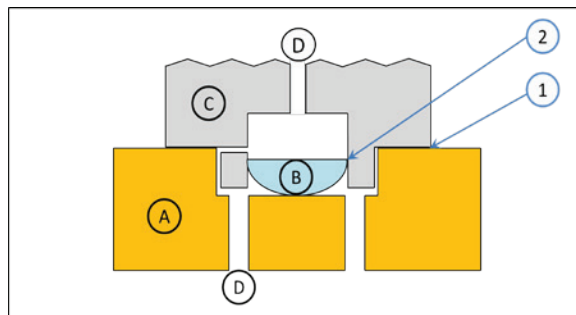


FIGURE 8: Semi-Kinematic mounting system. The brass puck, labelled A, has a diamond turned surface (1) that mounts against a diamond turned surface on the aluminum pedestal (C). A tungsten carbide half-sphere (B) is attached to the brass puck. A diamond turned cylindrical surface on the pedestal (2) mounts against the half-sphere. Holes for vacuum (D) are cut through both puck and pedestal for securing fragile parts to the pedestal.

CHARACTERIZATION

A power spectral density analysis of the system noise confirmed the quoted resolution of the sensors as well as indicated no significant drift for tests complete in fewer than 2 hours. This is important given the relatively slow assembly process and high alignment accuracy required for HEDS targets.

The power spectral density of the F/T sensor indicates white noise dominates frequencies above 10^{-2} and flicker noise dominates frequencies below. Thermal noise is not a concern when the manipulator is operated in the HED target fabrication machine shop, where the system will be used most frequently. This is an environmentally controlled facility so no thermal noise was expected.

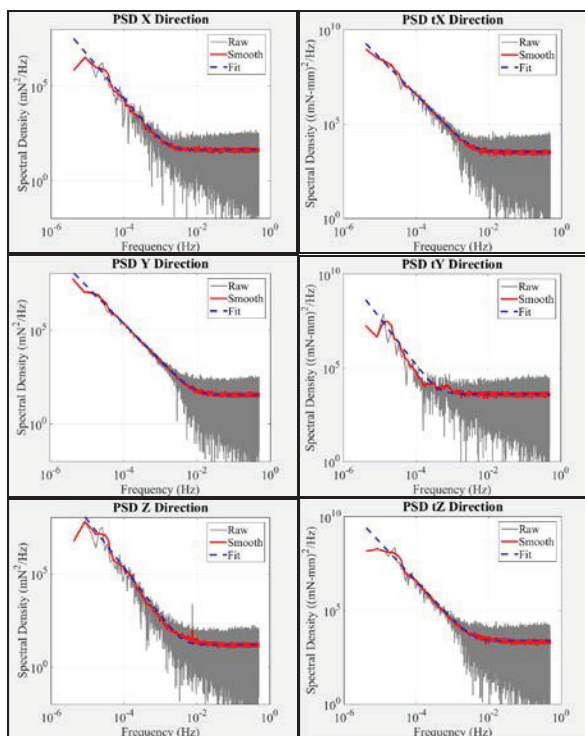


FIGURE 9: Power Spectral Density analysis of each axis (Force in X,Y,Z and Torque in θ_x , θ_y , θ_z) in the LLNL HED machine shop. White noise appears as the horizontal band at frequencies higher than about 10^{-2} . Flicker-like noise dominates frequencies lower than about 10^{-2} .

The expected noise (1 standard deviation) for a given period can be calculated from the power spectral density (PSD). The noise in the F/T sensor is fairly constant for measurements/tests that are completed within 30 minutes. The noise in the force has a standard deviation is approximately 3.92 mN (0.4 gf) and the torque shows a standard deviation between 29 and 44 mN-mm (3 and 4.5 gf-mm) depending on the axis. Both values are close to the quoted resolution of the F/T sensor. The standard deviation of the noise really increases after about 2.7 hours, indicating drift in the sensor.

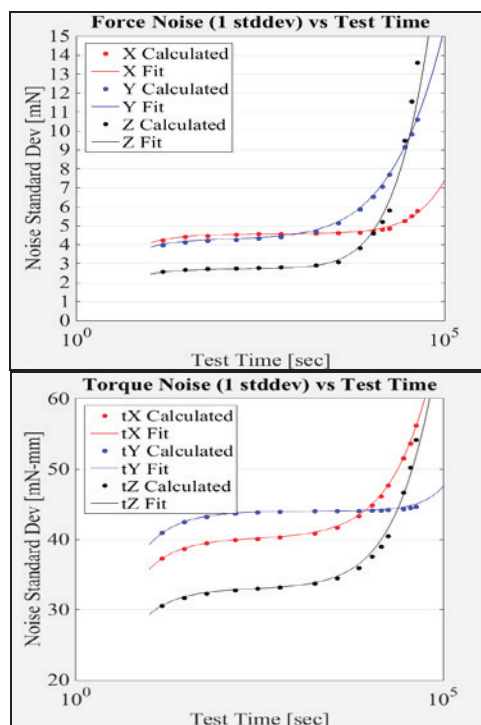


FIGURE 10: Noise vs time in F/T sensor. Noise in the sensor is fairly flat for time periods less than 30 minutes. After approximately 2.5 hours drift starts to significantly reduce the accuracy of the sensor.

The PSD of the displacement sensor was also calculated and showed similar results. White noise dominates regions at frequencies above 10^{-2} while flicker noise dominates frequencies below. A difference between the displacement and F/T data is a peak in the PSD around 10^{-5} . The displacement data was taken in the office environment above the machine shop which is not as tightly thermally controlled. The resulting data and PSD show a thermal noise source that corresponds to the day/night cycle (and thus temperature cycle) of the office space.

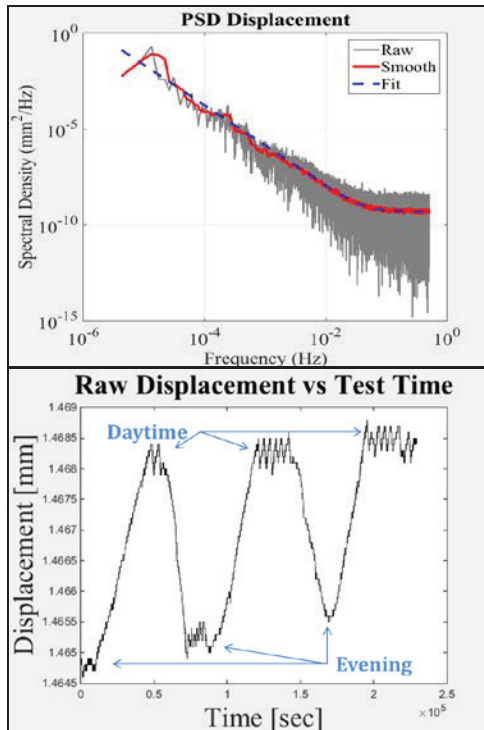


FIGURE 11: PSD and raw data from 2.5 day run of displacement sensor. White and flicker noise are dominant as expected. There is also a bump in the PSD around 12hrs, which corresponds to the day/night temperature variation in the office space where the sensor readings were taken.

The calculated standard deviation of the noise in the displacement sensor was calculated to be 0.05 μm , an order of magnitude less than the quoted accuracy of the sensor. The drift becomes significant around 2hrs. Combining the displacement and F/T sensor characterization indicates assembly or testing should be completed in less than 2 hours and the sensors should be zeroed between tests. In addition, a PSD should be run with the system in its final location to create an accurate assessment of the contributing noise sources and their impact.

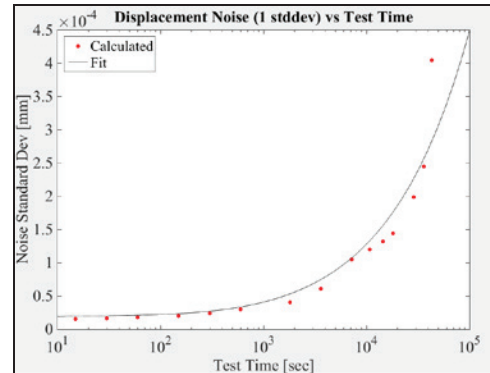


FIGURE 12: Noise vs time in the displacement sensor. The expected noise is less than 0.05 μm for tests lasting less than 2 hrs. Above 2hrs the sensor begins to drift significantly.

NANO INDENTER COMPARISON

The force and displacement sensors enable the manipulator to be used as a rough material characterization system in addition to an assembly system. To gauge the usefulness, ball indentation data was taken on JX-10 foam by the manipulator as well as on a commercial Nano indenter. The comparison is shown in Figure 13.

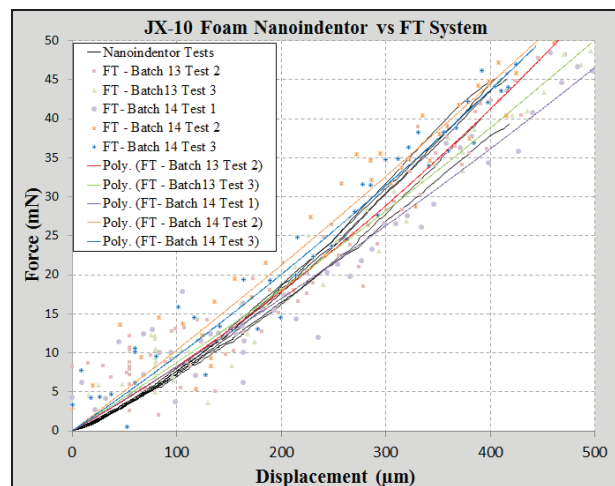


FIGURE 13: Comparison of Nano indenter and Manipulator Force/Displacement tests of JX-10 foam. A 1mm steel ball was mounted to a pedestal on the manipulator and actuated into the foam. The resulting data from the F/T and displacement sensors was compared to data taken by a Nano indenter.

To mimic a Nano indenter, a 1mm ball tip was mounted to the top stage of the manipulator. A JX-10 foam billet was mounted to the bottom

stage. The force and displacement were measured as the ball was pressed into the foam. A sample from the same batch was tested on a commercial Nano indenter. The data was compared to qualify the manipulator as a material indenter.

As a material indenter, the manipulator shows a similar force vs displacement trend as the commercial Nano indenter. The noise in the manipulator data is consistent with the characterization results. Even with the noise, the overall polynomial fit of the manipulator is a fairly good approximation of the Nano indenter results. The manipulator data overestimates the force at low displacements and appears to be trending a little low at high displacements. Using the manipulator as a material indenter will result in a good rough approximation of material properties, but it should not be used if high precision is needed.

CONCLUSIONS

The design and characterization is presented of a new device which provides precision manipulation for target fabrication. This manipulator combines kinematic coupling, precision motion stages and force/displacement feedback to facilitate accurate and precise assembly of high energy density science targets. The manipulator is shown to provide sub-micron position resolution, combined with mN-scale force resolution.

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REFERENCES

- [1] R. Hibbard and M. Bono, "An Overview of the Target Fabrication Operations at Lawrence Livermore National Laboratory," *ASPE Annual Meeting*, 2005.
- [2] M. Flaid, T. Plewa, P. Keiter, R. Drake, M. Grosskopf, C. Kuranz and H. Park, "Design of a supernova-relevant Rayleigh–Taylor experiment on the National Ignition Facility. I. Planar target design and diagnostics," *High Energy Density Physics*, vol. 12, pp. 35-45, 2014.

TUNABLE DYNAMICS PLATFORM FOR MILLING EXPERIMENTS

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INTRODUCTION

Time and frequency domain milling process models may be implemented to enable pre-process parameter selection for optimized performance [1]. To complete these simulations and validate the results, the system dynamics must be known. Typically, the cutting tool flexibility dominates the system dynamics, although the workpiece can introduce significant flexibility in some cases as well.

To realize a validation platform with simple (often single degree of freedom) dynamics, flexures are routinely used to support the workpiece; see Fig. 1. In this configuration, the flexure stiffness is selected to be much lower than the tool stiffness so that the tool dynamics can be effectively ignored. The experimental challenge with using flexures is that the damping is low. This can lead to unrealistic testing scenarios. In this paper, a flexure platform with tunable stiffness, natural frequency, and viscous damping is described which offers an ideal platform for milling process dynamics experiments.

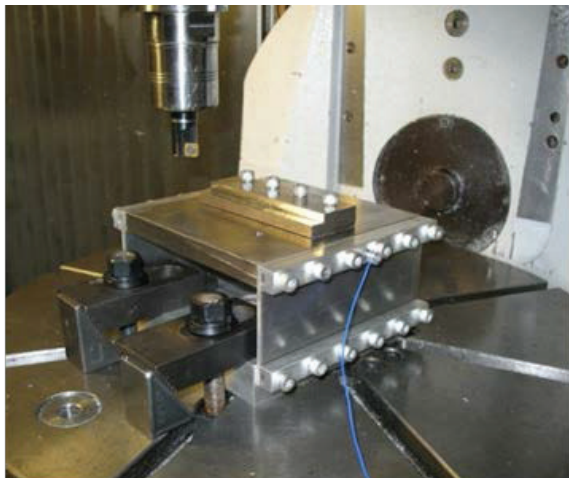
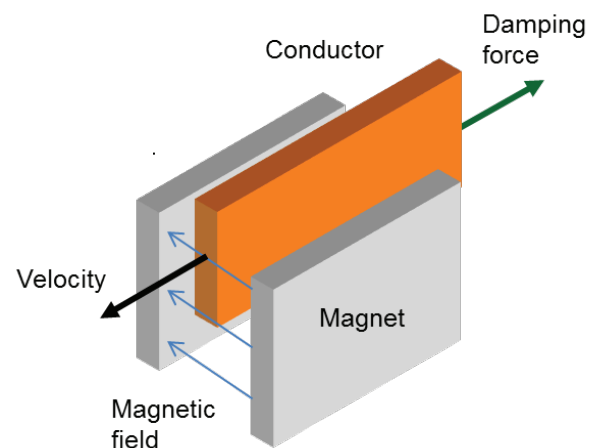


Figure 1. Parallelogram, leaf-type flexure used for milling experiments.

PLATFORM DESCRIPTION

The basis for the milling platform designed and tested in this study is a parallelogram, leaf-type flexure. By selecting the leaf geometry and workpiece/platform mass, the stiffness and natural frequency can be defined to meet the experimental requirements. The design approach is described in [2] and, for brevity, is not discussed here. As noted, however, the



damping for this geometry is low.

Figure 2. Schematic of an eddy current damper.

To introduce higher damping using a first principles model, the addition of an eddy current damper is proposed. The viscous (velocity-dependent) damping force for an eddy current damper can be described analytically. Figure 2 displays the motion of a conductor¹ relative to a magnet (or magnet pair) with the motion perpendicular to the magnet pole direction. The eddy current density, $\vec{J}$, depends on the conductivity, σ , and the cross product of the velocity, $\vec{v}$, and magnetic field, $\vec{B}$; see Eq. 1. The eddy current force is then calculated as the volume integral of the product of the eddy current density and the magnetic field; see Eq. 2. Mathematically, the two cross products yield a

¹ The conductor is a conductive, non-magnetic material. Aluminum and copper are common choices.

damping force which acts in the direction opposite to the velocity.

$$\vec{J} = \sigma(\vec{v} \times \vec{B}) \quad (1)$$

$$\vec{F} = \int_V (\vec{J} \times \vec{B}) dV \quad (2)$$

The damping force magnitude, F , is described by Eq. 3, where δ is the conductor thickness, B is the magnetic field strength, S is the magnet area, α_1 incorporates surface charge effects, α_2 incorporates end effects from the finite width conductor, and v is the velocity magnitude [3]. As shown, Eq. 3 can be rewritten as the product of a viscous damping coefficient, c , and the velocity magnitude. This viscous damping coefficient enables model-based damping prediction and selection for milling operations.

$$F = (\sigma \delta B^2 S (\alpha_1 + \alpha_2)) v = cv \quad (3)$$

The eddy current damper concept displayed in Fig. 2 was embedded inside an aluminum flexure as shown in Fig. 3. The conductor was attached to the flexure platform and two permanent magnet sets were attached to the flexure base, one on each side of the conductor. The arrangement of the eight 25.4 mm square magnets is shown in Fig. 4; aluminum screws were used to avoid disrupting the magnetic field.

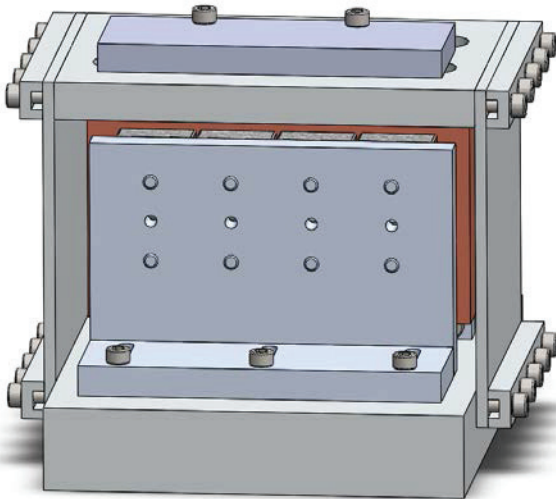


Figure 3: Flexure with embedded eddy current damper.

DAMPED FLEXURE DESIGN

The solid model displayed in Fig. 3 was constructed and tested. As shown in Eq. 3, the viscous damping coefficient can be predicted using σ , δ , B , S , and the edge effect terms, which serve to reduce the damping value; see Table 1. The conductor was a 19.1 mm thick copper plate that extended outside the magnet surface area. The magnetic field strength was measured at 64 locations over the surface of the eight permanent magnets using a gaussmeter (Integrity Design & Research Corp., IDR-329-T). Measurements were performed at a distance of 0.8 mm from the surface; this was the air gap between the magnets and conductor after assembly. The average value for all 64 measurements at the 0.8 mm distance is listed.

Table 1: Eddy current damper design parameters.

Parameter	Value
σ	5.96×10^7 A/V-m
δ	19.1 mm
B	4580 Gauss (0.458 T)
S	4.6×10^{-3} m ²
α_1^*	0.352
α_2^*	-0.177

*These terms were calculated using Eqs. 9 and 20, respectively, from [3].

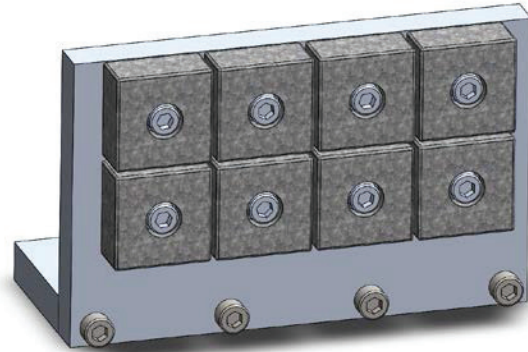


Figure 4: Permanent magnet mount.

For the values listed in Table 1, the predicted eddy current damper c value is 192 N-s/m. The corresponding dimensionless damping ratio, ζ , is calculated using Eq. 4, where k is the flexure stiffness provided in Eq. 5 and m is the equivalent flexure mass given by Eq. 6 [2]. In Eq. 5, E is the aluminum leaf's elastic modulus (69 GPa), w is the width (101.6 mm), t is the thickness (3.8 mm), and l is the length (88.9 mm). In Eq. 6, m_p is the combined mass of the

platform, conductor, top leaf clamps, and fasteners (2.098 kg) and m_l is the leaf mass (0.12 kg). The predicted damping ratio is 0.063 (6.3%). Using the flexure stiffness (1.093×10^6 N/m) and mass values (2.277 kg), the predicted undamped natural frequency is $f_n = 110$ Hz.

$$\zeta = \frac{c}{2\sqrt{km}} \quad (4)$$

$$k = 2Ew \left(\frac{t}{l} \right)^3 \quad (5)$$

$$m = m_p + 2 \left(\frac{26}{35} m_l \right) \quad (6)$$

EXPERIMENTAL RESULTS

Modal tests were performed to identify the actual damping ratio for the flexure. The setup is shown in Fig. 5. An instrumented hammer was used to excite the structure and the response was measured using a low-mass accelerometer. The modal parameters extracted from the single degree of freedom frequency response function were: $f_n = 110$ Hz, $k = 1.06 \times 10^6$ N/m, and $\zeta = 0.046$. The disagreement in the damping ratio is attributed to approximations in the charge and edge effects.

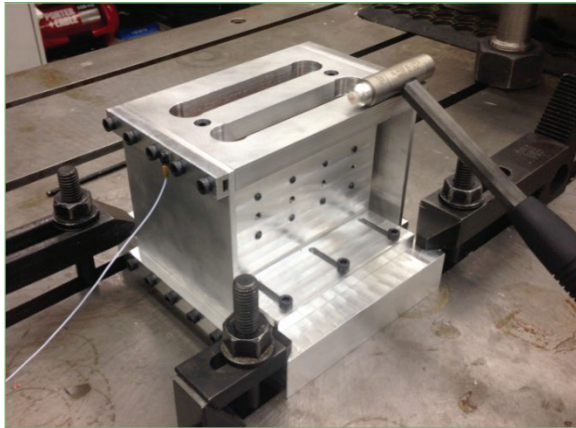


Figure 5: Impact testing setup for flexure.

Modal tests were also performed after removing the magnets so that the eddy current damping effect was eliminated, but the structure was not otherwise modified. The modal parameters extracted from the single degree of freedom frequency response function were: $f_n = 108$ Hz, $k = 9.03 \times 10^5$ N/m, and $\zeta = 0.014$. The addition of

the eddy current damper provided a 229% increase in damping.

Given the successful fabrication and testing of the flexure, machining trials were completed to demonstrate: 1) agreement between the predicted and experimental milling stability as a function of spindle speed and axial depth of cut; and 2) the change in the stability behavior with and without the eddy current damper in place.

First, a 6061-T6 workpiece was added to the platform and modal tests were repeated for the new setup as mounted on the CNC machine table; see Fig. 6, where the accelerometer attached to the flexure was used to identify stable and unstable cutting conditions. The new modal parameters were: $f_n = 110.6$ Hz, $k = 9.29 \times 10^5$ N/m, and $\zeta = 0.043$ with the magnets in place and: $f_n = 108$ Hz, $k = 9.03 \times 10^5$ N/m, and $\zeta = 0.015$ with no magnets.



Figure 6: Milling setup showing damped flexure, two-flute endmill, and accelerometer.

For the two-flute endmill/6061-T6 aluminum workpiece, the specific cutting force was 700 N/mm² and the force angle was 70 deg. Using these values together with the flexure dynamics (the tool point frequency response was an order of magnitude stiffer and was ignored), stability lobe diagrams were generated for the damped and undamped flexure setups [1]. Figure 7 shows the limiting axial depth of cut (b_{lim}) as a function of spindle speed (Ω) for a 25% radial immersion up milling cut.

Two spindle speeds were selected to validate the milling stability: 2000 rpm and 3500 rpm. In both cases, the predicted increase in the limiting depth of cut with the addition of the eddy current

damper is obtained. The results are presented in Table 2. Cuts were identified as stable or unstable based on the frequency content of the accelerometer signal, A. If significant content amplitude was observed at frequencies other than the tooth passing frequency and its harmonics, the cut was considered unstable. The chatter frequency was verified to be near the flexure natural frequency in each case.

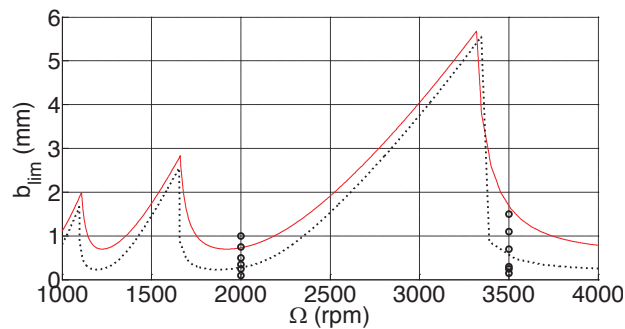


Figure 7: Stability limit with magnets (solid) and without magnets (dashed). The test points from Table 2 at 2000 rpm and 3500 rpm are marked by circles.

Table 2: Stability testing results. U indicates the undamped flexure and D indicates the damped flexure. Stable = o, unstable = x, marginal = m.

2000 rpm			3500 rpm		
b (mm)	U	D	b (mm)	U	D
0.1	o	o	0.15	o	o
0.25	m	o	0.25	m	o
0.35	x	o	0.3	x	o
0.5	x	o	0.7	x	o
0.75	x	m	1.1	x	o
1.0	x	x	1.5	x	x

Accelerometer frequency content examples are provided for the undamped flexure at a spindle speed of 3500 rpm in Fig. 8 (0.15 mm axial depth) and Fig. 9 (0.7 mm). In Fig. 8, content at the tooth passing frequency (116.7 Hz) and its multiples (as well as the runout frequencies) is observed. This cut is stable. In Fig. 9, on the other hand, the signal amplitude is 30 times larger with the dominant peak near the flexure natural frequency. This cut exhibited chatter.

CONCLUSIONS

This study described a new approach for prescribing the structural dynamics in machining stability testing. It demonstrated a passive platform with tunable dynamics that can be selected at the design stage. The design

process was detailed so that other researchers can implement the new strategy.

REFERENCES

- [1] Schmitz T, Smith, S (2009) *Machining Dynamics: Frequency Response to Improved Productivity*. Springer, New York, NY.
- [2] Smith, S (2000) *Flexures: Elements of Elastic Mechanisms*. CRC Press, London, UK.
- [3] Bae, JS, Moon, KK, Inman, D (2005) Vibration suppression of a cantilever beam using eddy current damper. *Journal of Sound and Vibration* 284:805-824.

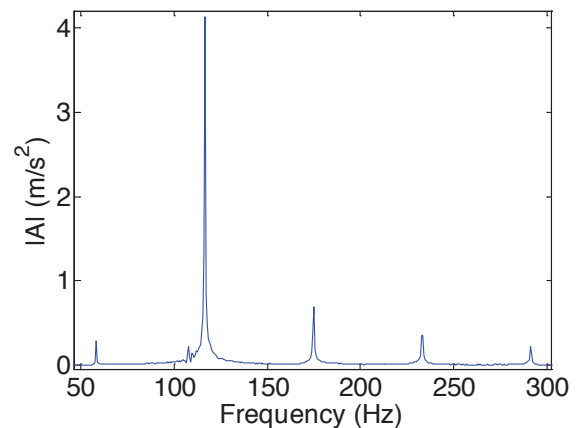


Figure 8: Stable cut at 0.15 mm axial depth and 3500 rpm spindle speed for the undamped flexure.

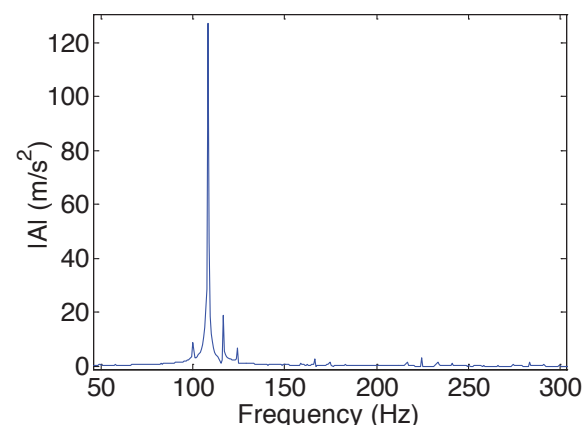


Figure 9: Unstable cut at 0.7 mm axial depth and 3500 rpm spindle speed for the undamped flexure.

Internal Stresses in Plastic Optical Components Caused by Injection Molding and Injection Compression Molding

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INTRODUCTION

This is a fundamental study on internal stresses in plastic optical components molded by injection molding (IM) and injection compression molding (ICM). The correlation of the formed internal stresses with processing parameters - specifically melt temperature, injection speed and packing pressure - are investigated and the results for both molding processes are compared [1].

The study is dedicated to the manufacture of plane optical specimens, using polymethyl-methacrylate (PMMA), cycle olefin copolymer (COC) and cycle olefin polymer (COP).

EXPERIMENTAL SET-UP

Both processes are conducted on an ARBURG ALLROUNDER 221K injection molding machine, which is equipped with a multi-cavity two-plate mold. Exchangeable cavity inserts are designed to form a round and plane molding with a wall thickness of 2 mm. The size of the cavities is identical for both IM and ICM. The tool inserts are fitted into the movable part of the mold as shown in Figure 1.

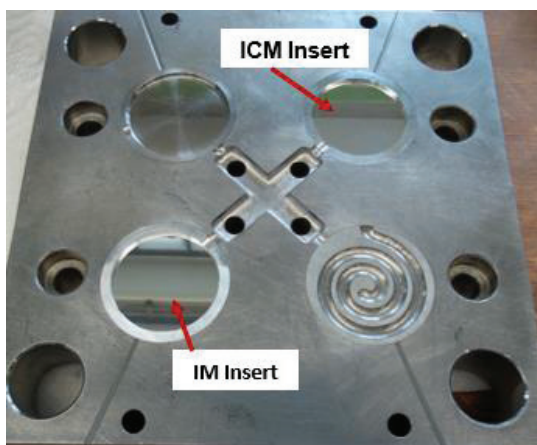


FIGURE 2. Exchangeable cavity inserts for ICM and IM fitted into the multi-cavity mold [1]

Both cavities are injected simultaneously to ensure identical processing parameters and environmental conditions.

The tool insert for IM is made up of two parts, namely a cylindrical stamp and a mounting. The stamp has a diameter of 40 mm and is fixed in the mold (cf. Fig. 2, left). The cavity insert for ICM comprises five parts: a mounting, a punching stamp, an internal guide rod, a pack of four parallel disc springs and two safety pins. The internal guide rod is fixed to the mold and the punching stamp is placed over it to allow for axial movements. The punching stamp is attached to the guide-rod with two pins to prevent it from being dislodged from its position during the process (cf. Fig. 2, right).

The method of compression in ICM can be described as a simultaneous opening and closing compression. A “breathing effect” on the axially moveable punching stamp is created within the closed mold by attaching the punching stamp to springs, whose movement is neither controlled nor regulated by the machine, but is instead the direct result of the cavity pressure brought about by the flow and shrinkage of plastic melt.

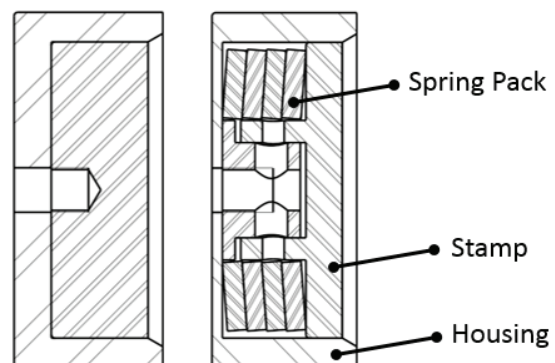


FIGURE 2. Exchangeable tool inserts for IM (left) and ICM (right) [1]

The fixed plate of the molding tool is equipped with two pressure sensors (*Kistler* type 6157B), recording the pressure in the cavity at the farthest point from the gate and simultaneously at a point just in front of the gate in the runner.

The internal stresses formed in the optical components are assessed with the photoelastic method by using a polariscope manufactured by *Optotechnik Kreuznach*, whose principle function is shown in Figure 3.

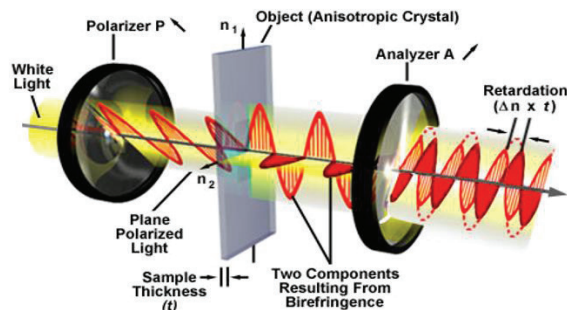


FIGURE 3. Principle function of a polariscope [2]

A polariscope reveals stressed areas as a series of multicolored bands or fringes. Each color is represented by a specific retardation, which is proportional to the degree of stress in the material at that point. [3]

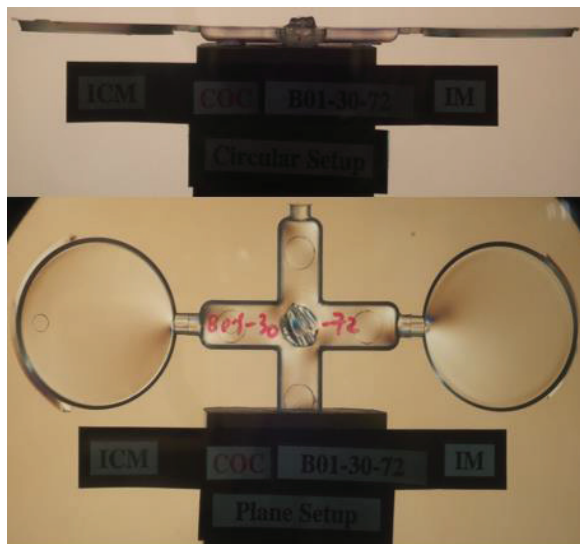


FIGURE 4. Parts molded by ICM (plane discs on the left) and IM (discs on the right) with COC, inspected sideways in a circular polarization setup (top) and from the top in a plane polarization setup (bottom) [1]

The specimens are assessed from the side with circular polarization and from the top by using a plane polarization setup (cf. Figure 4).

Black fringes known as isoclinics define the area with fixed directions of the main tension, while colored fringes known as isochromatics show the difference between the principle stresses as well as maximum shear stresses. Both isoclinics and isochromatics appear together in the plane polarization setup, but the isoclinics are broad and dark and hide the isochromatics, which appear as black lines, strands or areas. [4]

The circular polarization setup however eliminates the isoclinics and is used to observe the isochromatics. A Michel-Levy chart [5] is used to interpret the observed patterns. The stress uniformity in the specimens is determined by the number of color fringes observed, while the difference in magnitude of stresses is determined by the difference in fringe order or color as well as by the intensities of the colors.

EXPERIMENTAL RESULTS

The peak cavity pressures recorded in both IM and ICM are quite similar, but a significant difference is observed in the residual cavity pressure. When the plastic melt solidifies in IM, the residual cavity pressure dissipates due to shrinkage (cf. Fig. 5).

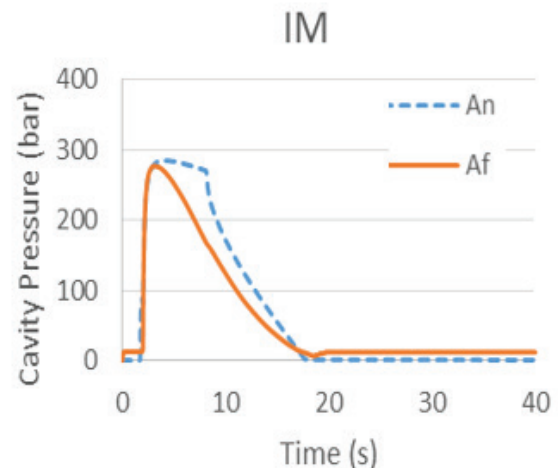


FIGURE 5. Cavity pressures recorded in the IM cavity. An: Cavity pressure near the gate; Af: Cavity pressure remote to the gate. [1]

In ICM however, the residual cavity pressure decreases to a certain point, but does not approach zero. This indicates that the movable punching stamp, which is attached to the spring pack, continues to actively compress the plastic molding even after it has solidified, resulting in a dynamic equilibrium. The cavity pressure drops to zero bar, when the part is demolded.

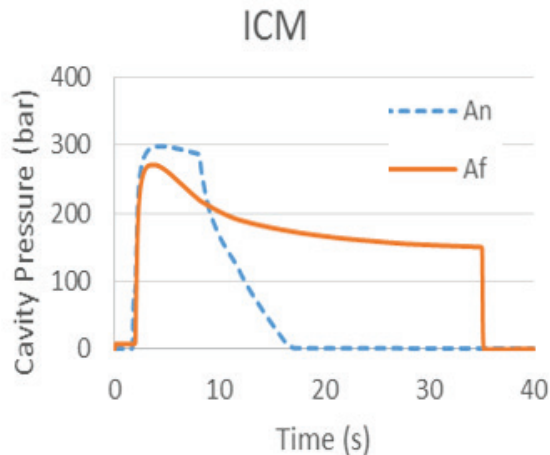


FIGURE 6. Cavity pressures recorded in IM cavity. An: Cavity pressure near the gate; Af: Cavity pressure remote to the gate [1].

Both IM and ICM are performed with the same basic settings of the processing parameters as given in Table 1. A packing time of six to eight seconds is chosen, depending on the material.

TABLE 1. Molding parameters used in the experiments shown in Fig. 5 and Fig. 6 [1].

Material	Tool Temp. [°C]	Melt Temp. [°C]	Filling Speed [ccm/s]	Packing Pressure [bar]
PMMA	90	240	30	500

Examining the effect of the processing parameters, internal stresses in both processes are found to be lower at higher melt temperatures and at

Examining the effect of the processing parameters, internal stresses in both processes are found to be lower at higher melt temperatures and at lower packing pressures, while injection speed shows very minimal to no effect. However, in the minimal change that is observed, it is found that internal stresses are lower at higher filling speeds [5].

Comparing the two processes, molding samples produced with ICM have lower internal stresses and a more uniform distribution of the stresses than parts produced by conventional injection molding (IM) (cf. Figure 7 & 8). Again, a stronger retardation relates to higher internal stresses.

It can also be concluded that the effect of using a different process (ICM or IM) on internal stresses is much more significant than the influence of the processing parameters.

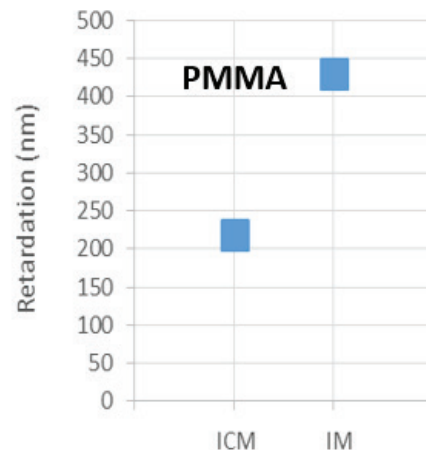


FIGURE 7. The effect of molding process on internal stresses for PMMA [1].

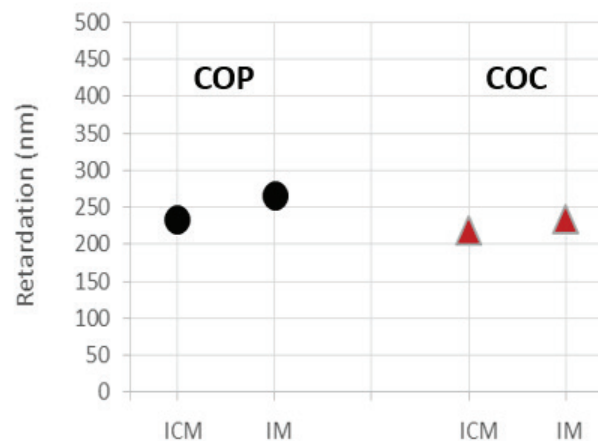


FIGURE 8. The effect of molding process on internal stresses for COP and COC [1].

Again, both IM and ICM were performed with the same set of processing parameters per material. Since COC has the lowest viscosity at processing temperatures, a lower packing pressure is applied.

TABLE 2. Molding parameters used in the experiments shown in Fig. 7 and Fig. 8 [1].

Material	Tool Temp. [°C]	Melt Temp. [°C]	Filling Speed [ccm/s]	Packing Pressure [bar]
PMMA	90	240	30	500
COC	100	250	30	300
COP	100	240	30	500

CONCLUSIONS

The effect of melt temperature on internal stresses can be explained by the stress relaxation phenomena. Polymer melts at higher temperatures require more time to cool and solidify, which consequently provides more time for stress relaxation. The longer the time available for stress relaxation, the smaller the residual stresses frozen in the molding. The lower viscosity of the melt due to the higher melt temperature also causes stress relaxation to occur more rapidly.

The slightly higher internal stresses observed at lower injection speeds are due to the orientation of the melt. A higher injection speed equates to a higher flow rate, which reduces the level of orientation in the molding. A lower orientation results to lower internal stresses.

The expansion of the frozen surface layer is also decreased at a higher flow rate, as the thermal convection becomes more dominant than thermal conduction to the colder mold wall, when the melt flows faster.

A higher packing pressure means that the deformation process in the material occurs under higher levels of stress, which results in higher orientation of the melt. Higher packing pressure also increases the constraining time of the melt. Although the packing time is held constant, higher amounts of melt are being pushed into the cavity to compensate shrinkage. A higher constraining time decreases the time available for stress relaxation. This in turn causes the frozen-in stresses to be higher.

The primary difference between the IM- and ICM process is the compression mechanism. Melt in IM is packed by implementing a packing pressure that pushes more melt into the cavity through the restricted gate before sealing.

Melt in ICM on the other hand experiences a uniform compression imposed by a punching stamp in addition to the packing pressure applied through the gate. Furthermore, the punching stamp in the ICM cavity is axially moveable, which allows for a change in the cavity volume during the filling and packing stages. The cavity volume in IM remains the same throughout the process. Therefore, it can be assumed, that the difference in stress level and uniformity observed in IM and ICM is due to the additional compression as well as volume adaptation by the axially moveable punching stamp in the chosen ICM setup with a spring-pack.

REFERENCES

- [1] Fadzil, L.: "Internal Stresses in Plastics Optical Components caused by Injection Molding and Injection Compression Molding". Master-Thesis, Beuth University of Applied Sciences, Berlin, 2015.
- [2] "Olympus Microscopy Resource Center," Olympus America Inc., 2012. [Online]. Available: <http://www.olympusmicro.com/primer/lightandcolor/birefringence.html>.
- [3] R. S. Sirohi: "Wave Optics and Its Applications", Orient Longman Limited, 1993.
- [4] P. Postawa and D. Kwiatkowski: "Residual stress distribution in injection molded parts," *Achievements in Materials and Manufacturing Engineering*, vol. 18, no. 1-2, pp. 171-174, 2006.
- [5] University of Cambridge, "University of Cambridge: Michel-Levy chart," [Online]. http://www.doitpoms.ac.uk/tlplib/optical-microscopy/michael_levy.php.

INFLUENCE OF POLYMER MELT PROPERTIES IN PRECISION INJECTION-MOLDING OF MICROFLUIDICAL AND OPTICAL PARTS

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INTRODUCTION

Today, a large variety of optical and microfluidical products is reproduced efficiently by injection molding techniques.

However, little is known about the influence of the melt rheology on the product quality in precision replication by injection molding. Fresnel grooves in molds for lenses or sidewalls of grooves in a cavity represent obstacles, which can change flow-speed and internal shearing conditions within the molten polymer.

In this paper, the influence of polymer melt properties and cavity surface structures in injection molding is investigated. The geometries of the replicated structures are designed to represent surfaces commonly found in optical and microfluidical applications. The mold-filling mechanisms are observed for the thermoplastic materials Polymethyl Methacrylate (PMMA), Cyclic-Olefinpolymer (COP) and Cyclic-Olefin-Copolymer (COC).

MELT PROPERTIES

Olefinpolymers find increasing use in optical and medical applications, due to advantageous properties as compared to other translucent thermoplastics, such as Polymethyl Methacrylate (PMMA). Olefinpolymers are less hygroscopic and may be used at higher temperatures, which is very favourable when replicating microfluidic chips or optical parts for rough environmental conditions. However, beyond melting temperature Olefinpolymers show a much lower viscosity in comparison to PMMA.

A standard testing device (cf. Fig. 1) is used in order to measure the Melt Flow Rate (MFR), which is a very common indicator for polymer melt properties. The higher the MFR at a given temperature and extrusion weight, the lower the viscosity of the melt.

Thermoplastic feedstock is molten in a heated vertical cylinder and kept at a controlled, preset temperature. The melt is extruded through a die of a given diameter by means of a piston, which is pressed downward by a defined weight. Then, the mass of the extrusion is repeatedly measured for a given span of time until a defined volume is squeezed out (cf. Fig. 1).

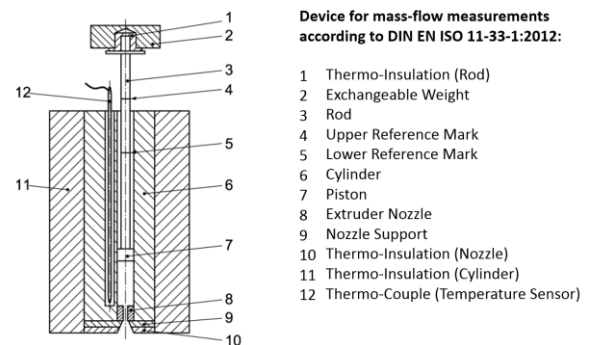


FIGURE 1. Device for melt flow measurements according to DIN EN ISO 11-33-1:2012

MFR measurements (cf. Fig. 2) show that the viscosity of both Olefinpolymers is much lower as compared to PMMA at typical processing temperatures, ranging from 220° to 275° Celsius [1].

In this case, PMMA has to be loaded with a higher weight on the piston in order to achieve enough extrusion for measurements. The MFR is directly influenced by the melt temperature and the impact of an increasing temperature is most significant for COC.

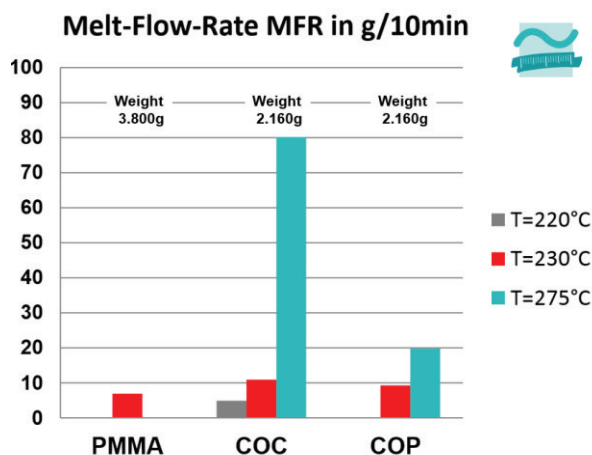


FIGURE 2. MFR for Cyclic-Olefin-Copolymer (COC), Cyclic-Olefin-Polymer (COP) and Polymethyl Methacrylate (PMMA) [1]

INJECTION MOLDING SETUP AND PARAMETERS

Injection molding experiments are carried out on an “Arburg Allrounder 221k” screw-type injection molding machine with a cold-runner and pre-heated tool, using Cyclic-Olefin-Copolymer (COC), Cyclic-Olefin-Polymer (COP) and Polymethyl Methacrylate (PMMA).



FIGURE 3. Injection-Molding Machine Arburg Allrounder 221k

The movable side of the tool holds two cavities for exchangeable cylindrical tool inserts with a diameter of 40 mm. In these experiments, one cavity is always equipped with an insert, which produces a plane disc with optical surface quality, while the other cavity holds an insert with geometric structures.

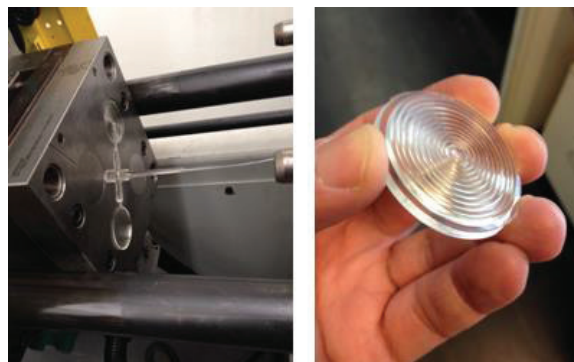


FIGURE 4. Open Tool with two cavities after molding (left) and single part (right)

Due to their low viscosity at processing temperatures, Olefinpolymers require carefully adjusted molding parameters, such as melt- and tool temperature, filling speed and packing pressure.

Table 1 gives an overview of parameter settings, which were found to be advantageous for producing a shot-weight of 14 g on a conventional, screw-type molding machine with a screw-diameter of 25 mm. Weighing and visual inspection of the parts after molding yielded a suitable packing time of 5 seconds, regardless of the material [1].

TABLE 1. Advantageous injection molding parameters.

Material	Tool Temp. [°C]	Nozzle Temp. [°C]	Filling Speed [ccm/s]	Packing Pressure [bar]
PMMA	80	240	20	650
COC	100	245	20	300
COP	100	245	20	350

MELT FLOW ON SURFACES WITH GEOMETRIC STRUCTURES

The melt flow in cavities with structures of different geometries is observed by filling studies, where the injection is stopped before the mold is filled completely. Thus, the polymer is frozen while filling the cavity.

These filling studies are carried out with two structured tool inserts. The first tool is a Fresnel-type insert with fairly large concentric grooves with a triangular cross-section of 1 x 1.5 mm. The second insert features an etched grating of parallel grooves with a depth of 50 μm .

Replication of Concentric Grooves

Simultaneous injection molding of the Fresnel-type part and a plane cylindrical disc reveals that geometric structures partially slow down the molten polymer and eventually split the melt stream into two separate flow-fronts, which reunite when the mold is filled completely while the holding pressure is applied [2].

One flow-front progresses along the plane back of the cavity, while the other flow-front spreads in a semi-circular way along the Fresnel-type grooves on either side from the gate. When the polymer is frozen before the cavity is filled completely, the v-shaped secondary flow-front is clearly visible (cf. Fig. 5).

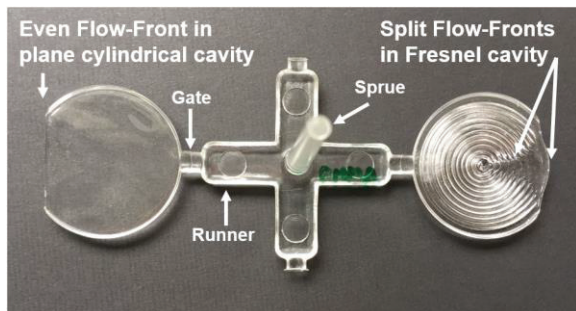


FIGURE 5. Injection Molded part (PMMA) with sprue and runner. Cavity filled to approx. 80% of max. filling volume.

Splitting of the flow-front is observed for all three thermoplastic materials. Filling studies with COP show similar results (cf. Fig. 6). Eventually, the flow fronts reunite in one particular spot. This may lead to deviations from the desired geometry, such as imperfectly filled structures.

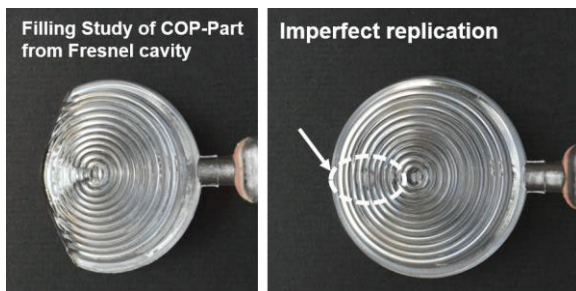


FIGURE 6. Filling study with split flow-fronts in COP (left) and imperfect replication in COP (right). Both parts on runner and slightly tilted for better visibility.

Moreover, weld lines in the surface and subsurface are to be expected, where the flow of molten polymer converges [3,4]. Weld lines appear as microscopic grooves in the surface and inhomogeneous material is commonly observed underneath the weld-line. Therefore it is to be expected that optical properties are also inhomogeneous, when flow-fronts meet within the functional zones of an optical part.

Replication of Parallel Grooves

In order to observe the melt-flow on submillimeter-structures, a grating of parallel grooves is etched into the polished surface of a cylindrical insert by lithography. The material of the insert is highly pure E-copper (Cu-ETP) [5].

The depth of the grooves is $50\text{ }\mu\text{m}$ at a groove width of 0.45 mm and a spacing of 0.8 mm as resolved by a *Perthometer* tactile stylus profiler in Fig. 7. The dimensions of the grooves were chosen similar to those often found in microfluidic applications (e.g. Lab-on-a-Chip systems for bio-medical studies on cell growth).

The insert can be rotated on a central screw. The orientation of the grooves towards the gate and the incoming melt is indicated by the angel φ according to Fig. 7. At an orientation of $\varphi=0^\circ$, the grooves are parallel to the gate and at $\varphi=90^\circ$ the grating is perpendicular to the gate.

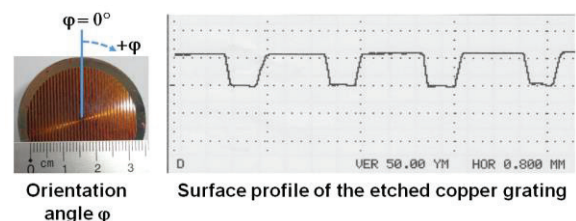


FIGURE 7. Etched copper grating and orientation angle of the grooves towards the gate. Tool insert (left) and profile (right) [5].

When PMMA is injected, a continuously curved single flow-front progresses through the cavity, as would be generally expected for a laminar flow originating from a single gate (cf. Fig. 8).

With the much less viscous COC however, the melt does not spread evenly. The central part of the flow-front, which initially entered the mold first, is dragged behind as the melt bulges to the left and right, curving outwards at the flanks and inwards towards the center.

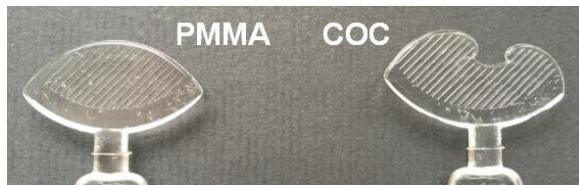


FIGURE 8. Filling study on tool insert with etched grating, using PMMA (left) and COC (right).

Rotation of the grating alters the shape of the flow-front (cf. Fig. 9). Again, the central part of the melt is dragged behind and eventually outflanked by secondary flows of melt. Apparently, contact with the grating alters the melt flow in a way, which is currently not predictable [5].

Further studies will be dedicated to examine, if the observed melt-flow patterns have an impact on product quality (e.g. geometry, internal stresses, optical properties).



FIGURE 9. Filling study on tool insert with etched grating, using COC and various orientation angles.

CONCLUSIONS

Periodic structures commonly used in optical or microfluidical parts influence the melt-flow in injection molding. This may lead to unexpected splitting and joining of flow-fronts within functional areas of the part. Joining flow-fronts lead to weld-lines [3,4] and locally to inhomogeneous material properties [4].

Until now, the observed melt flow behavior cannot be predicted sufficiently. Further experiments will be conducted in order to develop mathematical models for the fluid- and solidification mechanics in cavities with microscopic structures.

Moreover, dedicated numerical models are required, which offer a local resolution (e.g. mesh-size) much finer than the lateral dimensions of the geometries to be replicated. In further studies, a numerical model is to be developed.

REFERENCES

- [1] Schmuetz, J.; Burak, S.; Jakob, T.: „Wie anspruchsvolle Polymere erfolgreich verarbeitet werden“. Maschinenmarkt 40/2014. Vogel Business Media, Würzburg.
- [2] Schmuetz, J.; Sivri, B.; Jakob, T.: „Spritzgießen optischer Bauteile aus Cyloolefinpolymeren und –copolymeren“. Proc. Research Day 2015, Beuth Hochschule für Technik, 04/2015.
- [3] Tosselo, G.: „Precision Moulding of Polymer Micro Components“. PHD Thesis DTU Mechanical Engineering, Copenhagen, Denmark, 2008.
- [4] Nguyen-Chung, T.: „Strömungsanalyse der Bindenahftformation beim Spritzgießen von thermoplastischen Kunststoffen“. Dr.-Ing. Thesis, Technische Universität Chemnitz, 2001.
- [5] Heinrich, T.: „Fließverhalten thermoplastischer Kunststoffschmelzen an mikro-optischen und mikrofluidischen Strukturen“. Master-Thesis, Beuth University of Applied Sciences, Berlin, 2015.

FUNDAMENTAL STUDY ON METAL FILM COATING BY LARGE-AREA EB IRRADIATION

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INTRODUCTION

In large-area electron beam (EB) irradiation method developed recently by our research group, high energy density EB can be obtained without focusing the beam, by an Explosive Electron Emission phenomenon [1]. Therefore, large-area EB of 60mm in diameter with almost uniform energy density distribution can be used for instant melting and evaporation of the material surface. It was clarified that high efficient surface smoothing of metal molds made of steel, or cemented carbide was possible [2,3]. Then, the large-area EB irradiation equipment has been already introduced into the market and this method is practically applied in the production site. Also this method can be applied to the surface finishing of biomaterials, such as stainless steels and titanium alloy [4].

On the other hand, many surfacing processes, such as coating and surface modifying have been widely applied to the industrial products to improve the surface functions including surface hardness, wear resistance, corrosion resistance, friction coefficient, which leads to high value-added productions, the lifetime extension of products, and resource saving. Also in die and mold making, surface coatings of thin layer with high hardness, high wear resistance and high corrosion resistance are often done after the surface finishing for long life of metal molds.

During the large-area EB irradiation, plasma generates above the substrate surface because of its high energy density of EB. This phenomenon would cause the sputtering of target material set near the substrate surface. If the target material deposits well on the substrate surface simultaneously with the melting and resolidification of substrate surface, the adhesion between the deposited film and the

substrate surface would be very strong. Also if the film coating is possible in the same equipment after the surface finishing by the large-area EB irradiation, a high efficient continuous surface processing of surface finishing and coating will be realized.

In this study, the substrate surface component and structure after the large-area EB irradiation using some kinds of metal target were investigated in order to discuss the possibility of large-area EB irradiation as a new metal film coating method. Furthermore, the effect of EB conditions on the film coating characteristics was discussed.

EXPERIMENTAL PROCEDURE

Fig.1 shows the experimental set-up for examining the possibility of metal film coating by large-area EB irradiation. The workpiece material is alloy tool steel SKD11 (in JIS) used widely as a metal mold. Metal tube was used as target, and put on the workpiece surface. The inner diameter of the tube is 16mm, and the outer 18mm, since the primary experiments showed that the inner diameter of about 15-20mm was optimal to form the film with uniform thickness. A jig holder block made of SKD11

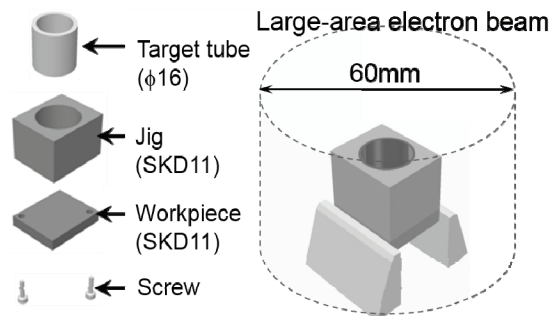


FIGURE 1. Experimental setup

was put on the workpiece plate to keep the target metal tube from moving with the impact of EB irradiation. Large-area EB is irradiated from above, as shown in the figure. In this study, two types of metal tube target were used. One is pure titanium and another is pure nickel.

As shown below, titanium and nickel films could be successfully formed by the large-area EB irradiation. The expected coating mechanism in this method is shown in Fig.2. Considering the fact that the plasma generates just on the substrate surface during EB irradiation, sputter deposition probably occur in the following way. First, when the large-area EB is irradiated to the substrate surface from above, the surface is rapidly heated up with high energy density of EB, and the surface material rapidly evaporates. Furthermore, the electrons in the beam collide with the generated vapor molecules. Consequently, the plasma is generated from the vapor. Next, some of ions in the generated plasma collide with the internal wall surface of target metal tube, and the metal is sputtered. The plasma generates inside the tube, which restricts the plasma expansion only to the inside and leads to efficient sputtering. Then, the sputtered metal deposits to the substrate surface. At this time, it is guessed that the substrate surface is melting by the large-area EB irradiation. Therefore, some of sputtered metal may be mixed with melted substrate material on the surface. Due to this mechanism, a strong adhesion of the deposited film with the substrate surface is highly expected.

NICKEL FILM COATING

In order to check the possibility of nickel coating by large-area EB, SEM observation and energy dispersive X-ray spectroscopy (EDX) analysis of the substrate surface were firstly undertaken. Fig.3 shows the results. The EB conditions were $7\text{J}/\text{cm}^2$ and 30shots, which are optimum condition for surface smoothing of SKD11 substrate surface. As shown in the figure, the substrate surface is smooth. It was confirmed that the surface roughness after the large-area EB irradiation was almost same or a little smaller, compared with that of ground surface before irradiation. Furthermore, nickel element can be detected and it is uniformly distributed on the surface after EB irradiation.

Next, the elemental content of substrate surface and the effects of EB irradiating condition were discussed. Fig.4 shows the variations of

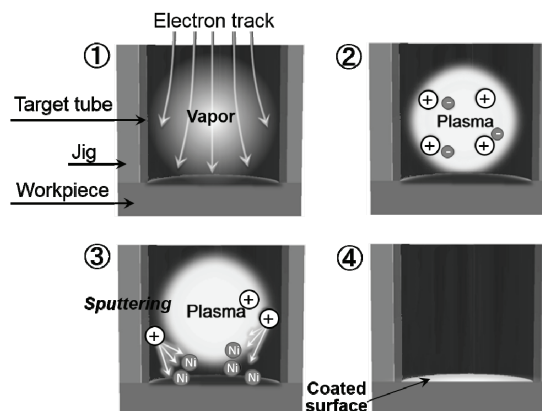


FIGURE 2. Expected coating mechanism

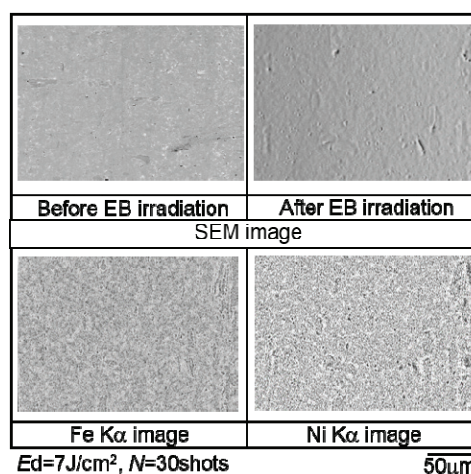


FIGURE 3. SEM images and EDX mapping analysis of nickel coated surface

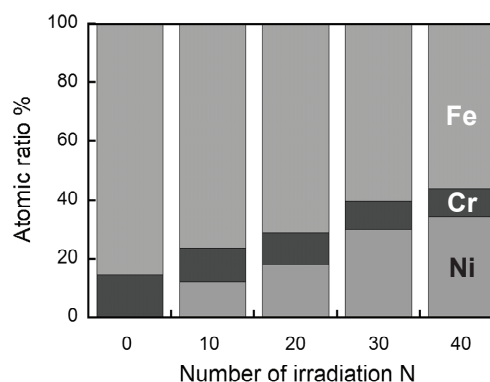


FIGURE 4. Elemental analysis results of nickel coated surface

elemental contents by EDX analysis with the number of EB irradiation under $15\text{J}/\text{cm}^2$ when a nickel tube was used as a target. The nickel content increases with EB irradiation. When the number of EB irradiation is 40shots, the nickel

content reaches 30%. On the other hand, the ratio of chromium to iron hardly changes with the number of irradiation, and keeps the original ratio of substrate material. In addition, it was confirmed that the nickel content increased with increasing the energy density of large-area EB. Therefore, it is considered that the film thickness would be controlled by these large-area EB conditions and sufficient thick nickel film can be formed on the substrate surface with large number of EB irradiation or large energy density.

Fig.5 shows TEM and each element images of the cross-section of nickel coated film on the substrate surface by large-area EB with nickel target. The border just below carbon layer deposited for the preparation of TEM observation is the film surface. Almost uniform nickel film of 500nm in thickness can be formed by EB irradiation. Also, the coated film surface is very smooth. In addition, at the interface between nickel film and substrate, nickel element seems to diffuse into the substrate. Furthermore, large grain of chromium carbide cannot be detected near the surface, which indicates that substrate surface was melted by EB irradiation and resolidified. The thickness is about 1.0 micron or thicker. From these results, strong adhesion of the nickel coated film to the substrate surface is highly expected, although tribological examinations may be needed.

Fig.6 is the XRD spectra of the surface before and after large-area EB irradiation. In the case before EB irradiation, the material structure of SKD11 can be detected clearly. On the other hand, nickel and some nickel alloys are detected on the surface after irradiation. The formation of nickel alloy indicates that the substrate surface material and nickel sputtered from the target somewhat mixed at atomic-level when the substrate material resolidified. This agrees well with the TEM observation and element image analysis results shown above.

In order to investigate the corrosion resistance of the nickel coated surface, anode polarization current density was measured using an electrochemical analysis system. Anodic polarization current is measured when the voltage between the workpiece surface and counter electrode increases at a constant potential scan rate. As an electrolyte, 3% sodium chloride solution was used. Fig.7 shows the anodic polarization current density curves of the substrate surfaces. For comparison, the

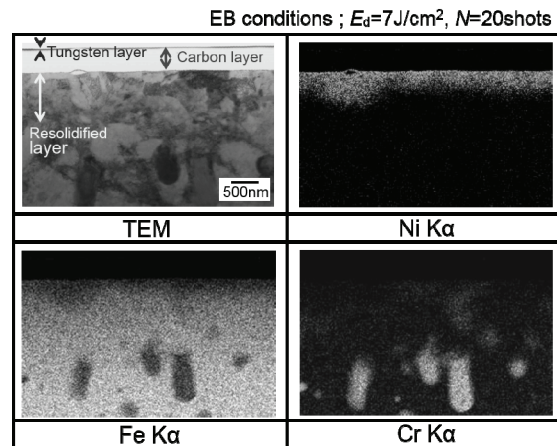


FIGURE 5. TEM and element images of cross section of coated film

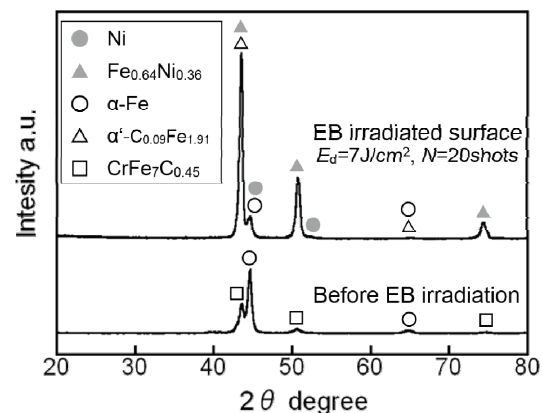


FIGURE 6. XRD spectra of surfaces before and after EB irradiation with nickel target tube.

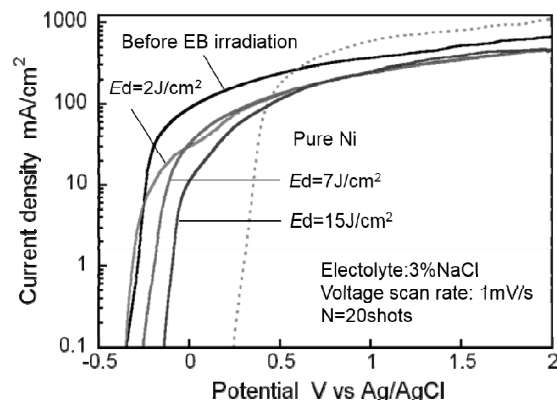


FIGURE 7. Anodic polarization current density curves

current density for pure nickel is also shown. The equilibrium potential for pure nickel is larger than those of other surfaces, which indicate its high corrosion resistance. The current density

for EB irradiated surfaces are smaller than non EB surface at any potentials. Therefore, the corrosion resistance of metal mold surface can be improved by this coating method with large-area EB. It is also noticed that the equilibrium potential becomes larger with the energy density of EB. This is probably because the nickel content and the nickel coated film increases with the energy density.

TITANIUM FILM COATING

The possibility of titanium film coating was investigated here. Fig.8 shows SEM and EDX mapping images of the substrate surface after large-area EB conditions with using a titanium target tube. Similarly to the nickel coating shown above, the substrate surface after EB irradiation is smooth and titanium element is distributed uniformly on the surface. Fig.9 shows the variations of elemental contents by EDX analysis with the number of EB irradiation when a titanium tube was used as a target. The titanium ratio on the surface increases with the number of EB irradiation. When the number of EB irradiation is sufficiently large, the ratio exceed 40%. Therefore, thick titanium film could be formed on the substrate surface by the large-area EB conditions.

XRD spectra of the surface before and after large-area EB irradiation with titanium target tube is shown in Fig.10. As shown in the figure titanium and some titanium alloys are detected on the surface after EB irradiation, apart from the original substrate structures. Also in the case of titanium target tube, the substrate surface material and titanium sputtered from the target somewhat mixed at atomic-level when the substrate material resolidified. The strong adhesion of the film to substrate surface can be expected.

REFERENCES

- [1] G. A. Mesyats, *Explosive Electron Emission*, URO-Press. Ekaterinburg, 1988.
- [2] Y. Uno, A. Okada, K. Uemura, P. Raharjo, T. Furukawa, K. Karato, *High Efficiency Finishing Process for Metal Mold by Large-area Electron Beam Irradiation*. Precision Engineering 29, 449-455, 2005.
- [3] Y. Uno, A. Okada, K. Uemura, P. Raharjo, *Method of Surface Treatment of a Die by Electron Beam Irradiation and the Treated Die*, US Patent 0183743, 2003.
- [4] A. Okada, Y. Uno, K. Uemura, P. Raharjo, J. A. McGeough, *Surface Modification for*

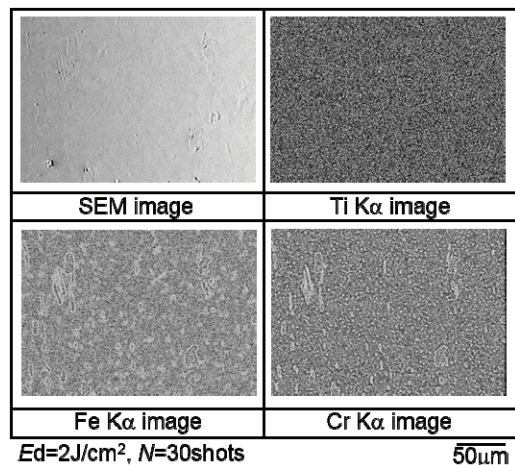


FIGURE 8. SEM images and EDX mapping analysis of titanium coated surface

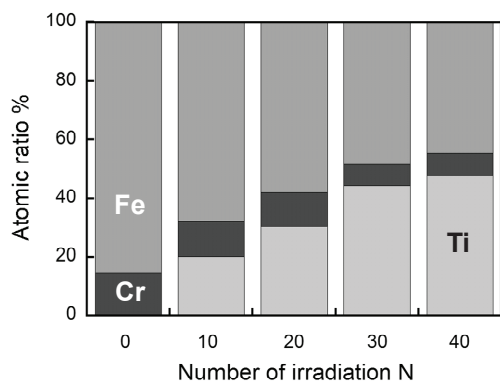


FIGURE 9. Difference in element ratio with number of EB irradiation

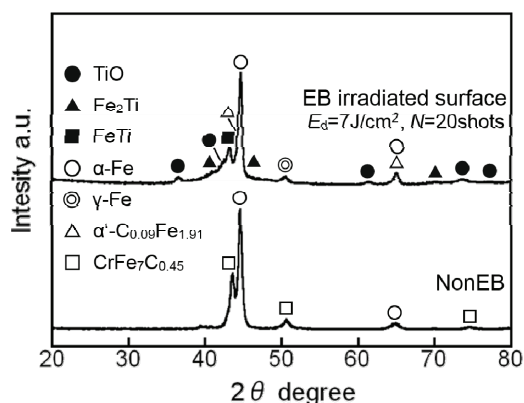


FIGURE 10. XRD spectra of surfaces before and after EB irradiation with titanium tube.

Orthopaedic Titanium Alloy by Wide-area Electron Beam, IMechE J. of Engineering Manufacture 221 B2, 173-178, 2007.

DESIGN AND ANALYSIS OF MEMS CARTWHEEL HINGE JOINT

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ABSTRACT

This paper presents the design and analysis of Micro-electro-mechanical systems (MEMS) flexural cartwheel hinge joints. MEMS provides a significant advantage of a relatively small footprint and flexural hinge joints have the improved performance in positioning compared to traditional joints. In this paper, we present a MEMS cartwheel hinge joint design. Finite Element (FE) model is built to obtain the compliance matrix. Based on the FE model, we conduct the structure-based performance analysis of the hinge joint. Furthermore, two novel cartwheel joints are proposed with the improved performance.

INTRODUCTION

Flexural hinge joints are widely used in precision mechanism design [1, 2, 3, 4] and they have the advantages of reduced wear, high precision, no back lash[5], reduced weight and maintenance. Cartwheel hinge joint is adopted as a revolute joint in compliant mechanism because of its higher out-of-plane stiffness and lower rotational in-plane stiffness.

A lot of research has been done in the design and modeling of cartwheel hinge joint. Kang and Gweon [6] built an analytical model for analyzing cartwheel hinge. Compared with Finite Element (FE) simulations, the errors is less than 10 %. Su et al. [7, 8, 9] built an analytical model based on the Screw Theory to analyze the performance of the cartwheel hinge joint. Pei et al. [10] derived two pseudo-rigid-body models to analyze the stiffness, compliance and motion of cartwheel flexural hinges. However, there is less research done in the cartwheel hinge joints of Micro-electro-mechanical systems (MEMS) [11, 12].

In this paper, we firstly present the design and fabrication of a MEMS cartwheel hinge joint. FE model is built for the structural analysis. In the FE

model, the joint is fixed at one end. A kinematic boundary condition constrained in 6 degrees of freedom (DOF) is placed on the other end. Based on the FE model, we derived the compliance matrix. Furthermore, two novel cartwheel hinge joint designs are proposed based on the modification of the original cartwheel joint. Based on the FE models again, we analyze the stiffness and compare them to the original cartwheel joint. The modified cartwheel joints have the improved performance in the different applications.

DESIGN AND FABRICATION OF A MEMS CARTWHEEL HINGE JOINT

A MEMS cartwheel hinge joint is presented in Figure 1. This hinge joint is fabricated by deep reactive ion etching (DRIE). By means of DRIE, the thickness of the beam can be fabricated to several hundred micrometers so that the flexural joints have a large out-of-plane stiffness. However, the thickness also depended on the width of the feature because DRIE fabrication is limited by the ratio of the feature dimension to the thickness.

In order to have a survivability during the fabrication and application, the beam should be designed to have a width as larger as possible. However, this increases the beam stiffness and further reduces the rotational performance of the cartwheel hinge joint. On the contrary, the width of the feature has the limit down because the fabrication has the limitations in the smallest feature which can be fabricated and the ratio of the feature dimension to the wall height. Here, we use a wafer with a 30 μm device layer, so the width of the beams are designed around 3-7 μm . The width of the cartwheel joint in Figure 1 is around 3 μm .

The cartwheel joint is composed of four beams: two top beams and two bottom beams. They are firstly connected in parallel respectively. Then,

the two top and two bottom beams are connected in serial. Compared to a single beam-based joint, the parallel connection of the beams increases the out-of-plane stiffness and survivability in the fabrication.

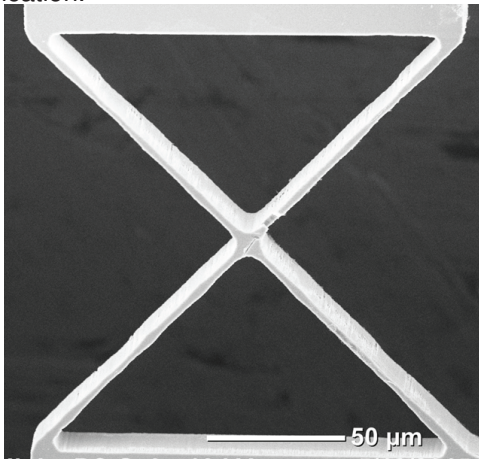


FIGURE 1. Design and fabrication of a MEMS cartwheel hinge joint

FINITE ELEMENT MODEL

In this section, we analyze the cartwheel joint design presented in Figure 1 by means of FE analysis in Figure 2. According to the dimensions measured in the previous section, we firstly build a 3D model in SolidWorks. Then, we input the 3D model into the FE simulation software. By means of this method, the simulation of a single joint is corresponding to the fabricated structure.

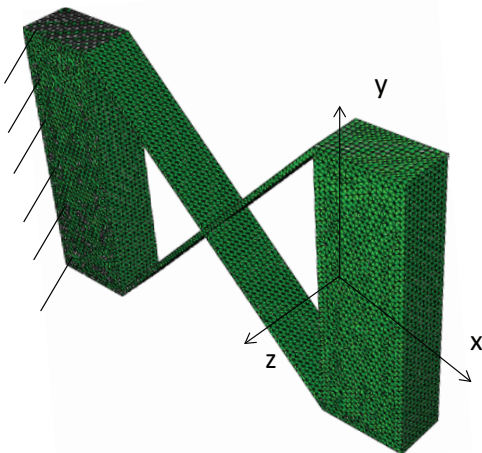


FIGURE 2. FE model of the cartwheel hinge joint.

In the FE model, one end of the joint is encastered. At the other end, we build the coordinate

system of the joint shown in Figure 2. A kinematic constraint of six degrees of freedom (DOF) is built on the plane where the coordinate is located. The origin of the coordinate system is the control point and the slave surface is the rectangular plane where the x axis and the y axis are located. At the origin of the coordinate, the forces and moments along the three axes are applied respectively. By means of this setting, the plane of the free end moves as a rigid plane when the loading is located at the origin of the coordinate. As shown in Figure 2, the joint is meshed with 124207 tetrahedral elements. Table 1 presents the translational and the rotational displacements of the origin corresponding to the forces and moments applied on the origin respectively. The subscript denotes the directions of the forces and the moments. They are the same directions as the axes of the coordinates.

When we use the cartwheel hinge joint as a revolute joint in kinematics, we want a larger in-plane rotational angle θ_z , but θ_x , θ_y , δ_x , δ_y , δ_z are smaller. As shown in Table 1, θ_z is 3.28E-03 radians when M_z is 1E-3 $N\mu m$ for this fabricated MEMS cartwheel joint. In the view of the structure, we want the cartwheel has the lower stiffness of M_z to θ_z . The displacements caused by F_y is usually coupled with M_z in the application of a revolute joint. The cartwheel joint can also be used in the other applications, such as a flexural compressible structure. In this application, we want a pure translational displacement δ_x , but θ_x , θ_y , θ_z , δ_y , δ_z are smaller. δ_x is 1.99E-1 μm when F_x is 1E-2 N for this fabricated MEMS cartwheel joint. In order to evaluate the performance of the cartwheel, we can also derive the compliance matrix of the cartwheel from Table 1. The compliance matrix is obtained by normalizing the displacements of 6DOF to the unit forces 1 N and the moments 1 $N\mu m$. The compliance matrix of the hinge joint shown in Eq. 1.

In this matrix, the third, forth, and fifth columns denote the out-of-plane stiffness of the joint. The third column is respect to the force F_z . The forth and fifth columns are respect to M_x and M_y respectively. 2.02E+03 denotes the translational displacement in the z direction caused by F_z with the unit $\mu m/N\mu m$. 7.31E-02 denotes the rotational angle in the x direction caused by M_x with the unit $\mu m/N$. 2.59E-01 denotes the rotational angle in the y direction caused by M_y with the unit $\mu m/N$. We can realize that the rotational compli-

TABLE 1. Translational and rotational displacements of the cartwheel hinge joint in FE Model.

	δ_x	δ_y	δ_z	θ_x	θ_y	θ_z
$F_x(1E-2 \text{ N})$	1.99E-01	5.27E-03	-2.63E-04	-1.04E-07	2.32E-06	-3.54E-05
$F_y(1E-5 \text{ N})$	-2.58E-04	1.87E-01	5.11E-06	-1.18E-08	-7.59E-08	2.48E-03
$F_z(1E-4 \text{ N})$	-1.62E-04	-9.77E-05	2.02E-01	2.06E-05	-1.95E-03	-1.24E-06
$M_x(1E-2 \text{ N}\mu\text{m})$	-3.77E-06	-2.35E-04	2.06E-03	7.31E-04	-2.70E-05	-3.10E-06
$M_y(1E-2 \text{ N}\mu\text{m})$	-1.71E-04	-2.19E-04	-1.95E-01	-2.70E-05	2.59E-03	-2.87E-06
$M_z(1E-3 \text{ N}\mu\text{m})$	-4.72E-04	2.48E-01	6.66E-06	-1.74E-09	-8.92E-08	3.28E-03

$$[C^c] = \begin{bmatrix} 1.99E+01 & -2.58E+01 & -1.62E+00 & -3.77E-04 & -1.71E-02 & -4.72E-01 \\ 5.27E-01 & 1.87E+04 & -9.77E-01 & -2.35E-02 & -2.19E-02 & 2.48E+02 \\ -2.63E-02 & 5.11E-01 & 2.02E+03 & 2.06E-01 & -1.95E+01 & 6.66E-03 \\ -1.04E-05 & -1.18E-03 & 2.06E-01 & 7.31E-02 & -2.70E-03 & -1.74E-06 \\ 2.32E-04 & -7.59E-03 & -1.95E+01 & -2.70E-03 & 2.59E-01 & -8.92E-05 \\ -3.54E-03 & 2.48E+02 & -1.24E-02 & -3.10E-04 & -2.87E-04 & 3.28E+00 \end{bmatrix} \quad (1)$$

ance of the y axis is higher than that of the x axis. This means the joint is easier to rotate along the y axis than the x axis. Meanwhile, this means the joint is more sensitive to the loading which cause the rotation along the y axis than along the x axis in the application of a revolute joint. However, these values are much smaller than $3.28 \mu\text{m}/\text{N}$, which denotes the rotation along the z axis by the moment along the z axis.

TWO MODIFIED CARTWHEEL JOINTS

In this section, we propose two novel cartwheel joints shown in Figure 3 and 4. The beams of the cartwheel hinge joints are curved in two directions.

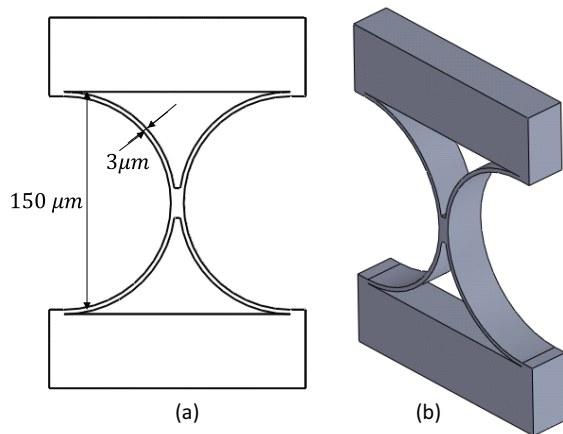


FIGURE 3. Modified cartwheel hinge joint A.

In order to conduct the structural analysis and compare it to the original cartwheel structure, we

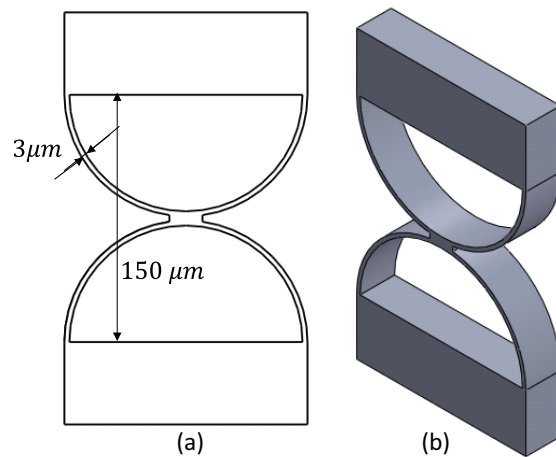


FIGURE 4. Modified cartwheel hinge joint B.

build the FE models shown in Figure 5 and Figure 6. The FE models are built with the same boundary conditions as the previous FE model. The same loadings are applied to the joints. The joint type A is modeled with 133616 elements in Figure 5 and the joint type B is modeled with 118267 elements in Figure 6. The results of the FE simulation are shown in Table 2. We can find that the type A has a 23 % larger rotational angle θ_z corresponding to the moment M_z than that angle of the original cartwheel joint. This means that the type A has the improved performance to be used as a revolute joint. However, type A has a lower out-of plane stiffness compared to the original design. Compared to the original joint, the

TABLE 2. Translational and rotational displacements of the modified cartwheel hinge joints in FE Model.

		δ_x	δ_y	δ_z	θ_x	θ_y	θ_z
Type A	$F_x(1\text{E-}2\text{ N})$	6.54E+00	2.05E-03	-5.64E-03	7.68E-05	1.03E-04	2.01E-05
	$F_y(1\text{E-}5\text{ N})$	1.88E-03	2.58E-01	-7.21E-04	-2.97E-07	8.15E-06	3.03E-03
	$F_z(1\text{E-}4\text{ N})$	-1.04E-02	-1.57E-03	1.70E+00	2.85E-06	-1.38E-02	-2.59E-05
	$M_x(1\text{E-}2\text{ N}\mu\text{m})$	1.58E-03	-5.06E-03	-7.79E-04	4.11E-03	1.00E-05	-7.33E-05
	$M_y(1\text{E-}2\text{ N}\mu\text{m})$	-7.93E-03	-1.21E-03	-1.38E+00	4.74E-06	1.84E-02	-2.36E-05
	$M_z(1\text{E-}3\text{ N}\mu\text{m})$	1.50E-03	3.03E-01	-7.43E-04	-9.39E-07	8.23E-06	4.04E-03
Type B	$F_x(1\text{E-}2\text{ N})$	5.12E+00	-6.08E-04	-1.18E-03	-2.11E-05	8.80E-06	1.45E-05
	$F_y(1\text{E-}5\text{ N})$	1.23E-03	1.05E-01	1.74E-04	1.60E-07	-2.17E-06	1.20E-03
	$F_z(1\text{E-}4\text{ N})$	-4.09E-03	6.50E-05	9.78E-01	1.42E-06	-1.14E-02	1.16E-06
	$M_x(1\text{E-}2\text{ N}\mu\text{m})$	1.29E-03	-3.54E-04	2.43E-04	9.03E-04	-2.99E-06	-3.04E-06
	$M_y(1\text{E-}2\text{ N}\mu\text{m})$	-7.66E-03	-1.01E-03	-1.14E+00	-1.18E-06	1.52E-02	-1.26E-05
	$M_z(1\text{E-}3\text{ N}\mu\text{m})$	7.66E-04	1.20E-01	1.82E-04	3.07E-07	-2.40E-06	1.60E-03

type B joint has a worse performance to be used as a revolute joint. However, the type B has a large compliance of δ_x over F_x . This means that the type B can be converted to be a prismatic joint in the x direction within a small motion.

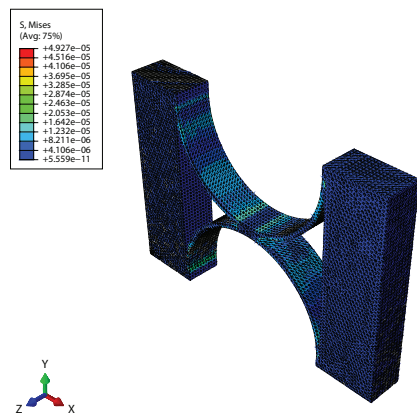


FIGURE 5. FE Model of the modified type A cartwheel hinge joint.

CONCLUSION

In this paper, we present the design and analysis of MEMS cartwheel hinge joints. The design and fabrication of a MEMS cartwheel hinge joint is illustrated. The FE model is built for the structural analysis and the derivation of the compliance matrix. Two modified cartwheel joints are proposed. Based on the FE models and the structural analysis, these two novel designs show the improved performances.

REFERENCES

- [1] Shi H, Su HJ, Dagalakis N, Kramar JA. Kinematic modeling and calibration of a flexure

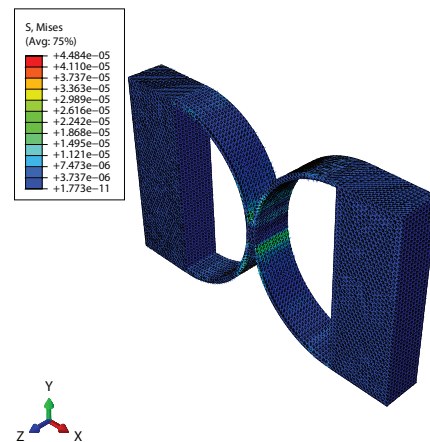


FIGURE 6. FE Model of the modified type B cartwheel hinge joint.

based hexapod nanopositioner. Precision Engineering. 2013;37(1):117 – 128.

- [2] Shi H, Su HJ. An Analytical Model for Calculating the Workspace of a Flexure Hexapod Nanopositioner. ASME Journal of Mechanisms and Robotics. 2013;5(4):041009.
- [3] Yang SH, Kim YS, Yoo JM, Dagalakis NG. Microelectromechanical systems based Stewart platform with sub-nano resolution. Applied Physics Letters. 2012;101(6):061909.
- [4] Hubbard NB, Howell LL. Design and characterization of a dual-stage, thermally actuated nanopositioner. Journal of Micromechanics and Microengineering. 2005 Aug;15(8):1482.

- [5] Shi H, Su HJ. Workspace of a Flexure Hexapod Nanopositioner. In: Proceedings of ASME IDETC/CIE; 2012. Chicago, Illinois, August 12-15.
- [6] Kang D, Gweon D. Analysis and design of a cartwheel-type flexure hinge. *Precision Engineering*. 2013;37(1):33 – 43.
- [7] Su HJ, Shi H, Yu J. Analytical Compliance Analysis and Synthesis of Flexure Mechanisms. In: Proceedings of ASME IDETC/CIE; 2011. Washington, DC, August 28-31.
- [8] Shi H. Modeling and Analysis of Compliant Mechanisms for Designing Nanopositioners. The Ohio State University. Columbus, OH, USA; 2013.
- [9] Su HJ, Shi H, Yu J. A Symbolic Formulation for Analytical Compliance Analysis and Synthesis of Flexure Mechanisms. *ASME Journal of Mechanical Design*. 2012;134(5):051009.
- [10] Pei X, Yu J, Zong G, Bi S, Su H. The modeling of cartwheel flexural hinges. *Mechanism and Machine Theory*. 2009;44(10):1900 – 1909.
- [11] Shi H, Su HJ, Dagalakakis N. A stiffness model for control and analysis of a MEMS hexapod nanopositioner. *Mechanism and Machine Theory*. 2014;80:246 – 264.
- [12] Shi H, Duan X, Su HJ. Optimization of the workspace of a MEMS hexapod nanopositioner using an adaptive genetic algorithm. In: *Robotics and Automation (ICRA)*, 2014 IEEE International Conference on; 2014. p. 4043–4048. Hong kong, May 31-June 7.

Authors Index

Aangenent, Wouter 136
Aarts, Ronald Godfried Karien Marie 95
Abed, Ovadia 52
Abolbashari, Mehrdad 389
Aguilar, Alejandro 567
Aguirre, Gorka 538
Ajay, Paras 177, 223
Allsop, John 15
Anton, Jayson 534
Aoyama, Hisayuki 317
Arneson, David A. 375, 379
Artoos, Kurt 303
Azari, Mohammad 389
Barber, Zeb 114
Beinhauer, Arne 273
Blackburn, Christopher 431
Boland, Richard J. 383
Bos, Arjo 131
Bosmans, Niels 469
Brasseur, Jason K. 461
Bridges, Phillip 474
Brinksmeier, Ekkard 58, 91, 273
Brouwer, Dannis Michel 95
Brown, Stephen B. 419, 465
Brown, Thomas G. 120
Butler, Sam 445
Calaon, Matteo 543
Cao, Z. C. 547
Capocasale, Christopher M. 321
Chao, Leon Shih 210
Chee, Sze Keat 280
Cheikh, Abdel Madjid 424
Chen, Shih-Chi 27, 86, 100
Chen, Yaolong 228, 311, 327
Cherala, Anshu 177
Cheung, Chi-Fai 491, 547
Choi, Hwan-Jin 245
Chou, Ta-Hsin 48
Clark, William H. 567

Collette, Christophe 303
Comley, Paul Douglas 15
Crouch, Stephen 114
Cuerden, Brian 194
Cullinan, Michael Arthur 183, 188, 480, 516
Dagalakis, Nicholas G. 369
Davies, Matthew A. 64, 440
Devitt, Andrew J. 218
DeWulf, Wim*
Doane, Danielle 577
Doergeloh, Timo 273
Doki, Kae 241, 299
Dong, Xin 148
Dow, Thomas A. 68, 200
Du, Jianjun 474
Duenner, Andrew 183, 480, 516
Dury, Martin Robert 419, 465
Eldredge, Jeff 336
Ellis, Jonathan D. 120, 403, 409, 415, 445
Elsborg, René 543
Fadzil, Liyana 587
Farahi, Faramarz 389
Felker, Sean J. 577
Ferrucci, Massimiliano*
Foremny, Eike 273
Forest, Craig R. 321
Fritz, Matthias 249
Fujii, Tatsuya 561
Fujisawa, Ryutaro 237
Furst, Stephen J. 200
Furutani, Katsushi 280
Garner, Sean 48
Garrard, Kenneth P. 200
Gatica, Anthony W. 232
Gawlik, Brian 42
Ghernaout, Amine 424
Gillich, Barbara J. 126
Gillmer, Steven R. 120, 415
Goetze, Kurt A. 534

***No Abstract Available**

Grigas, Michelle A.	52
Guo, Ping	331
Gweon, Dae-Gab	353
Habibi, Nasim	389
Hansen, Hans Norgaard	543
Harvey, Christopher	171
Hayakawa, Naoki	237
Hayashi, Akio	348
Head, Stephen T.	120
Hebrink, Timothy J.	32
Heinrich, Thomas	591
Henselmans, Rens	131
Hesler, Gregory S.	68
Higuchi, Toshiro	280
Hirakawa, Tetsuma	522
Ho, Lai-Ting	547
Holst, Gregory L.	321
Honda, Sakuro	435
Honeycutt, Andrew.	551, 583
Hopkins, Jonathan B.	166, 171, 555
Hosto, Jeffrey J.	126
Howard, Stephen Christopher	415
Hsieh, C. W.	48
Huang, H.	307
Huang, M. H.	48
Huber, Jeffrey M.	126
Huttenhuis, Stephan.	206
Imura, Ryousuke	457
Inoue, Motohiro	595
Inukai, Ryota	280
Ishibashi, Shingo.	241
Ito, Yusuke.	561
Jackson, Julie A.	171
Jacobs, Ralf	407
Jang, Sang-Don.	285
Je, Tae-Jin	245
Jeon, Eun-Chae.	245
Jeong, Jae-heon	353
Jeong, Ji Young.	245
Jeung, Sung-Hwa	218
Jia, Qian	228, 311, 327

Jiang, Zhu	357
Kafashi, Sajad	336
Kakiuchi, Toshifumi.	237
Kamiya, Kazuhide	394
Kang, Wenjun	474
Kato, Tomonori	522
Kaylor, Brant	114
Kearny, Steven	534
Kelly, Scott.	336
Kim, Hyoyoung	353
Kim, Yong Sik	526, 532, 599
Kiran, Kadir	571
Kivel, Edward J.	32
Kizaki, Toru	561
Knaapen, Raymond J.	19
Knapp, Byron R.	375
Kode, Sai Kumar	268
Kosters, Eric	407
Krishna, Shiva Ram.	285
Krishnan, Girish.	342
Kuhfuß, Bernd.	273
Kuo, K. T.	48
Kushnir, Emmanuil	567
LaBrake, Dwayne	42
Ladner, Ian	183, 188
Lan, Jie	311
Langelaar, Mathijs.	136
Lantz, Brian T.	160
Laro, Dick A. H.	136
Leach, Richard K.	110
Lee, ChaBum	398, 403
Lee, Hoon-Hee	485
Lee, Jae-Chang	485
Lee, Sun-Kyu	398
Lee, Vincent	126, 262
Łetocha, Aneta	503
Li, Fang	289
Li, Lei.	451
Li, Wei.	23
Li, X.	48
Li, Yuanda	321

Liebers, Melvin J.	379
Lin, Hung-Yih.	48
Lin, J. C.	48
Lin, S. M.	48
Liu, Liangliang.	228
Liu, Mingyu	491
Liu, Xiaoting	311
Lu, Chao	403
Ludwick, Stephen J.	293
Maier, Christian.	249
Martin, Hubert M.	194
Matichard, Fabrice	160, 303
Matsumoto, Kimihisa	394
McCarthy, Michael B.*	419, 465
McGruder, Dustin J.	321
Menezes, Joyson	571
Merghache, Sidi Mohammed	424
Miller, Jimmie A.	474
Mitsubishi, Mamoru	561
Mitsuyoshi, Yuta	241
Mittleman, Richard.	160
Mohammadi, Hossein	254
Morantz, Paul	15
Morgan, Christopher J.	37
Mototani, Suguru	241, 299
Muhamad, Mohd Ridha	258
Muralikrishnan, Bala	262, 431
Nagasaka, Kohei	348
Nagoshi, Masahiro	435
Nakao, Yohichi	348
Nayfeh, Samir Ali	105
Niehaus, Frank	206
Nomura, Takashi.	394
Okada, Akira	595
Okwudire, Chinedum E.	148, 154
Oss, Daniel D.	375, 379
Owen, Joseph D.	64
Panas, Cynthia	577
Panas, Robert Matthew	171, 555
Park, Eun-Suk	245
Park, Jaehyun	353

Park, Sang-Hyun	285
Patten, John A.	254
Pawlus, Pawel.	497, 503
Pérez de Nanclares, Aitor.	538
Phillips, Steven D.	126
Pratt, Jon R.	210
Preuss, Werner R.	58
Qian, Jun.	469
Rachakonda, Prem Kumar	262, 431
Radkins, Hugh.	160
Ransom, Tyler	583
Ravindra, Deepak R.	268
Reibel, Randy Ray.	114
Reizer Ph.D., Rafal	497, 503
Ren, Mingjun.	491
Rentsch, Rüdiger	58
Reynaerts, Dominiek	469
Ricci, Michael A.	445
Riemer, Oltmann.	58, 91, 273
Robertson, Norna Anne	160
Roos, Peter Aaron.	461
Rosielle, P. C. J. N.	131
Rubeo, Mark A.	80
Ruijl, Theo A. M.	142
Sakai, Yasunori	357
San Andres, Luis.	218
Sano, Sadao	595
Satheeshbabu, Sreeshankar.	342
Sawyer, Daniel S.	262, 431
Schlamming, Stephan	210
Schmidt, Peter	275
Schmitz, Tony L. . 64, 74, 80, 214, 403, 551, 571, 583	
Schmuetz, Joerg.	587, 591
Schönemann, Lars	91, 273
Senda, Taketo	317
Seugling, Richard M.	577
Shakarji, Craig M.	126, 262
Shapiro, Brett N.	160
Shaw, Jessica	577
She, Yu	526, 532
Shi, Hongliang.	363, 369

***No Abstract Available**

Shimada, Shoichi	435
Shinonaga, Togo	595
Shore, Paul	15
Shu, Deming	534
Shvyd'ko, Yuri	534
Simon, Michael D.	321
Singhal, Shrawan	52
Smith, R.	48
Smith, Stuart T.	336, 509
Smith, William L.	171
Sone, Shori	299
Song, Yuanping	555
Sreenivasan, S. V.	42, 52, 177, 223
Steinbuch, Maarten	131
Stevens, Ronald	407
Stockslager, Max A.	321
Stoupin, Stanislav	534
Sugita, Naohiko	561
Suit, Brandon	68
Sun, Guoao	188
Sun, Rongrong	451
Suzuki, Shinya	394
Takata, Masashi	595
Tamagawa, Tomoyuki	457
Tanaka, Tomohisa	357
Tarbutton, Joshua	398, 403
Tashiro, Hatsuzo	394
Theisen, Michael J.	120
Thorpe, Michael J.	461
Tobies, Hubert Florian.	249
Torii, Akihiro	241, 299
Tosello, G.	543
Troutman, John R.	64
Tseng, P. L.	48
Tsuchiya, Kensuke.	237
Tyler, Christopher	74
Uda, Yutaka	435
Uphaus, Timothy M.	171
Urata, Yoshiaki	394
Urreta, Harkaitz	538
Valdez, Lucas	275

van de Sande, Werner W. P. J.	95
van der Meulen, Stan	136
van der Veen, Gijs	136
van Keulen, Fred	136
Verdirame, Justin M.	105
Villarraga-Gómez, Herminso.	509
Walker, N.	15
Wang, Chen	409, 445
Wang, Chen-Yu	48
Wang, Dien	27, 86
Wang, Ji	100
Wang, K. G.	48
Wang, Tao	228
Watchi, Jennifer J.	303
Westfahl, Drew	223
Woodward, Sean D.	419
Woody, Shane C.	415
Wullms, Pieter.	142
Xiao, Weniei	307
Xin, Hao.	23
Yang, S.	48
Yang, Seung-Han.	485
Yano, Takeshi.	280
Yao, Tsung-Fu	516
Ye, Peiqing.	289
Yoon, Deokkyun	154
Yoshioka, Hayato	357
Yu, Xiangzhi	415
Yuasa, Yuki	561
Zha, Jun.	228, 327
Zhang, CongXue	327
Zhang, Dapeng	86
Zhang, Hui.	289
Zhang, Xiaowen	148
Zhao, G.	307
Zhou, Xi	27, 100
Ziegert, John C.	275
Zou, Yanhua	258

